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INVESTIGATIONS OF A VARIABLE GEOMETRY COMPRESSOR FOR A DIESEL ENGINE TURBOCHARGER

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by Thomas P. Oatway; James L. Harp, Jr.

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PROPULSION SYSTEMS LABORATORY

U.S. ARMY TANK AUTOMOTIVE COMMAND Warren, Michigan

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INVESTIGATIONS OF A VARIABLE GEOMETRY COMPRESSOR FOR A DIESEL ENGINE TURBOCHARGER

FINAL REPORT

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1.0 SUMMARY

An analytical and experimental investigation was conducted to determine the most practical method of surge control for advanced diesel turbocharger compressors. The results of the analysis indicated that the vaned diffuser actuation system represented the best method of achieving effective regulation over a broad range of flow rates.

A standard turbocharger compressor was selected for modification to variable diffuser geometry. A variable diffuser system was designed, fabricated, and incorporated in a turbocharger test rig. Bench tests indicated that the variable diffuser indeed had the ability to regulate flow and control surge without excessive reduction of efficiency or pressure ratio. A flow range of 3.5:1 was obtained at a pressure ratio of 2.4 between diffuser throat settings of 110% and 45% of the design value. Predicted efficiency at reduced vane settings was met or surpassed. Efficiency and pressure ratio at the design vane setting exactly matched the performance quoted for the production compressor, so no effect of variable diffuser geometry on peak compressor efficiency was indicated. Performance data generated at a larger than design vane throat area (110%) indicated higher efficiency potential than the design setting. Over a range of diffuser vane settings from 72% to over 110% of design (corresponding to a flow range of 2.6:1 at a pressure ratio of 2.4) less than 5% efficiency loss resulted. It is concluded that the variable diffuser compressor is an extremely useful approach for providing the more critical compressor range requirements of advanced diesel powerplants.

2.0 INTRODUCTION

2.1 Background

A trend toward increased turbocharger compressor pressure ratio is projected for future military diesel powerplants. The effect of higher design pressure ratio is known to reduce surge-free range and attainable efficiency of the compressor component. Unfortunately, attainment of adequate surge margin is already a problem with current highly turbocharged, high speed diesels, so future demands are likely to compound an already critical compressor-engine matching problem. The necessity of increasing surge range and maintaining reasonable compressor efficiency introduces a requirement for increased sophistication of the turbocharger compressor unit.

The nature of the compressor surge problem is illustrated in Figure 2.1.1, the compressor performance map for the turbocharger utilized on the Army VHO (525CID) diesel engine. Operating lines at constant engine speeds are superimposed to show that the location of the surge line limits the boost to low manifold pressures at reduced engine speeds. At the rated speed (2800 RPM) and at high compressor pressure ratios the operating line is far removed from the surge line, causing reduced compression efficiency. This is typical of the manner in which current turbocharged diesels are compromised in efficiency at the rated power point to obtain reasonable surge margin at low engine speeds. For advanced diesels it would be desirable to have a compressor provide roughly constant boost from the rated speed down through the mid-speed range, enabling the engine to develop its maximum torque at a low to medium speed (perhaps 1600 RPM).

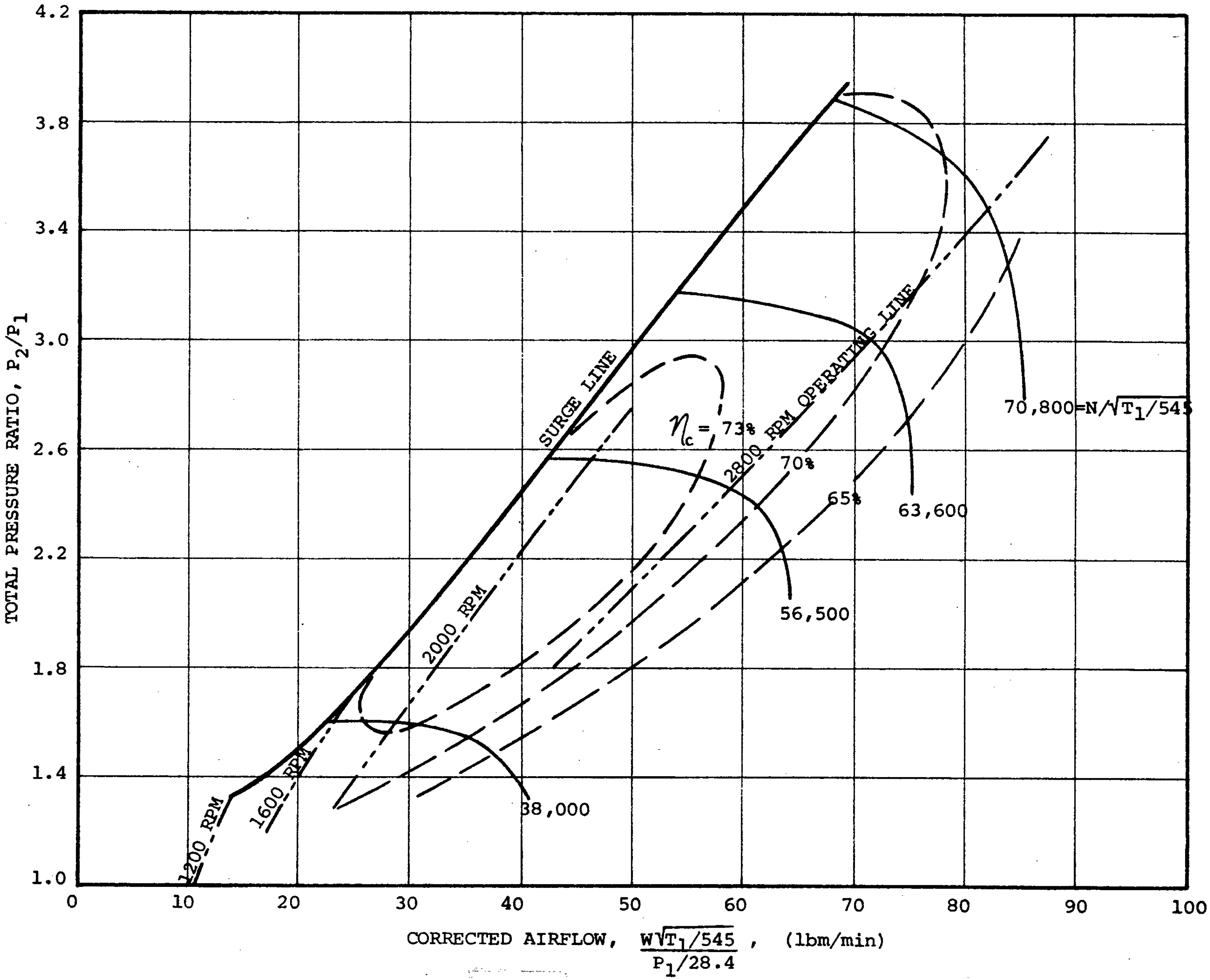


FIGURE 2.1.1. Standard Compressor Map with VHO Operating Line Superimposed.

2.2 Objective

The overall objective of this program was to analyze, design, and fabricate a variable geometry compressor for the VHO turbocharger which is capable of eliminating the problem of compressor surge. From an engine standpoint, the purpose of developing a surge-free compressor was to enable diesel power to be increased, and smoke reduced at low and medium speeds, yet maintain full engine power at high speeds. The specific objectives included:

1. Determine the most practical and least complicated method of adapting variable geometry to each major component of the compressor, including the inlet, diffuser, and rotor.
2. Analyze a variable geometry turbocharger compressor to determine the best probable combination of surge margin, pressure ratio, and efficiency at each operating point.
3. Analyze the possibility of applying aerodynamic methods as a means of achieving variable geometry performance with less mechanical complexity or higher reliability.
4. Evaluate special configurations of rotor blades and diffuser vanes and determine the compressor range availability.
5. Analyze the potential of a Pelton wheel hydraulic turbine supplement to the turbine drive of a turbocharger compressor compared to other promising part speed boost augmentation systems for diesel turbochargers.
6. Analyze the potential of valve timing to permit better low speed performance and less smoke while maintaining acceptable high speed performance.
7. Design two types of variable geometry compressor-

turbine units to the layout stage and recommend the preferred concept for fabrication.

8. Fabricate one variable geometry compressor for the VHO high efficiency turbocharger.

3.0 ANALYSIS

3.1 Surge Control

Three mechanical variable geometry methods have been employed on centrifugal turbomachinery to improve performance and increase operating range:

1. Variable impeller blades.
2. Variable inlet guide vanes.
3. Variable diffuser.

Variable blade techniques have been utilized in very large, low-speed, hydraulic power generating machinery (Kaplan turbines an example). Unfortunately, flow regulation by altering the pitch of turbocharger impeller blades is impractical because of extremely high rotational speeds and resulting structural limitations. Therefore, primary emphasis was given to the analysis of variable inlet guide vanes (IGV) and variable stators or diffusers as mechanical surge control methods for turbocharger compressors. Workable designs for both of these systems have been developed and put into production. The following sections will describe the capability of variable inlet vanes and variable diffusers for control of surge in turbocharger compressors.

3.1.1 Variable Inlet Guide Vanes

In this system of flow regulation vanes are placed in the compressor inlet section to give the incoming air a whirl component; prewhirl in the direction of rotation is called positive prewhirl and prewhirl counter to impeller rotation is termed negative prewhirl. If these vanes are fully variable the relative velocity vector approaching the rotor blade can be controlled to eliminate the tendency of rotor blades to stall (positive incidence) as flow rate through the machine is reduced. Thus, the contribution of the inducer to actual compressor surge is effectively eliminated.

In addition to alignment of relative air flow vectors with rotor blade geometry, a stage designed with a certain amount of positive prewhirl has a reduced inlet relative air velocity or Mach number entering the rotor (W_1) compared to a stage without prewhirl or with negative prewhirl. Therefore, even a stage with stationary inlet vanes has improved stability because reducing the relative approach Mach number tends to make the inducer less sensitive to varying incidence and allows the rotor to operate over a wider flow range before stall or gross separation of the blade leading edge suction surface takes place.

Figure 3.1.1 (a) and (b) represent the impeller inlet and exit velocity diagrams for a stage with positive and negative prewhirl. From the Euler work equation, the theoretical amount of work imparted to each pound of air as it passes through the impeller is given by:

$$E = \frac{U_2 V_{u_2} - U_1 V_{u_1}}{g_c}$$

Where E = Work per lbm of air

g_c = Gravitational Constant

U_2 = Impeller peripheral velocity

U_1 = Inducer velocity at mean radial station

V_{u_2} = Absolute tangential velocity at impeller exit

V_{u_1} = Absolute tangential velocity at impeller inlet

It can be seen from this equation that positive prewhirl reduces the theoretical work of the stage by an amount equal to $U_1 V_{u_1}/g_c$. Because of this loss of work the pressure rise of the stage at a given wheel speed is reduced, so even though there is a reduction of the minimum flow rate that the rotor can pass without stalling, the accompanying head rise loss limits the improvement in surge margin to small changes. In effect, varying prewhirl at a constant compressor speed is shown by tests (Ref. 1) to be very similar to merely changing compressor speed on a compressor without

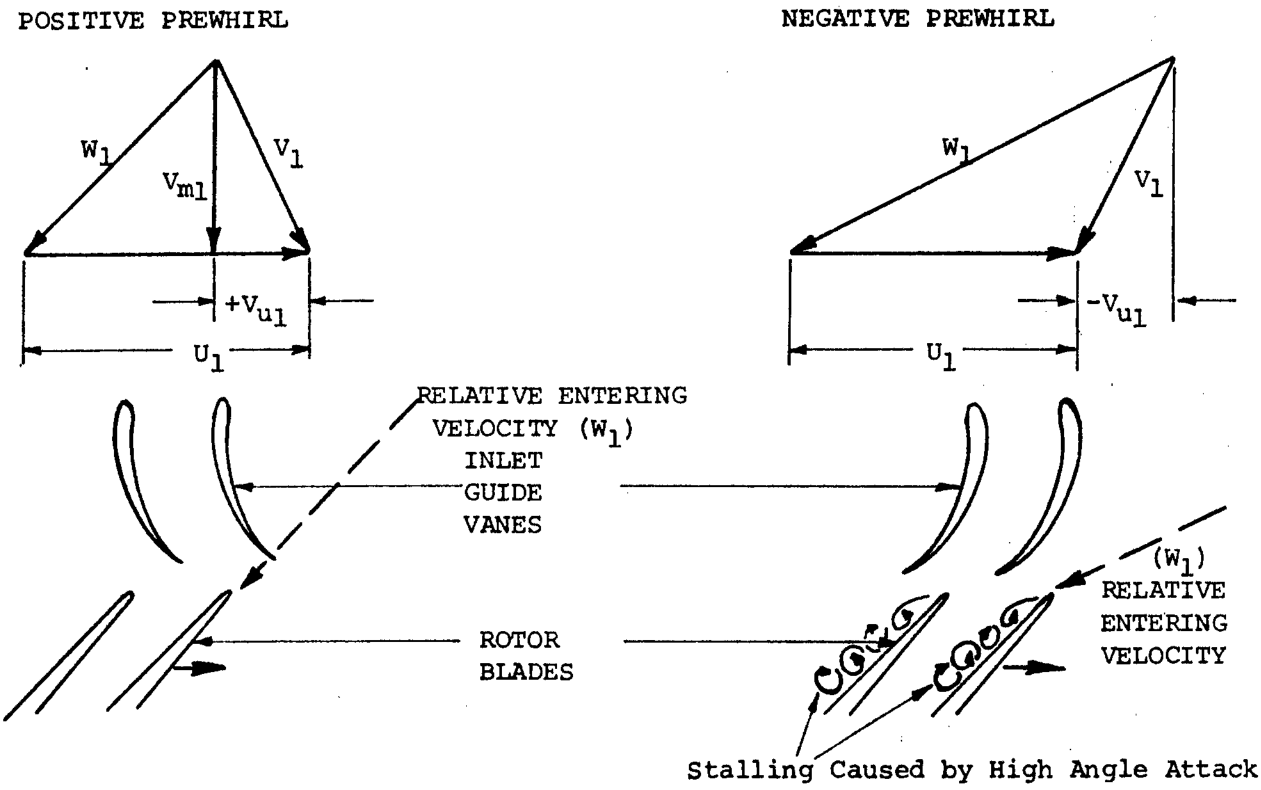


FIGURE 3.1.1(a). Inlet velocity diagrams for a radial compressor with positive or negative prewhirl.

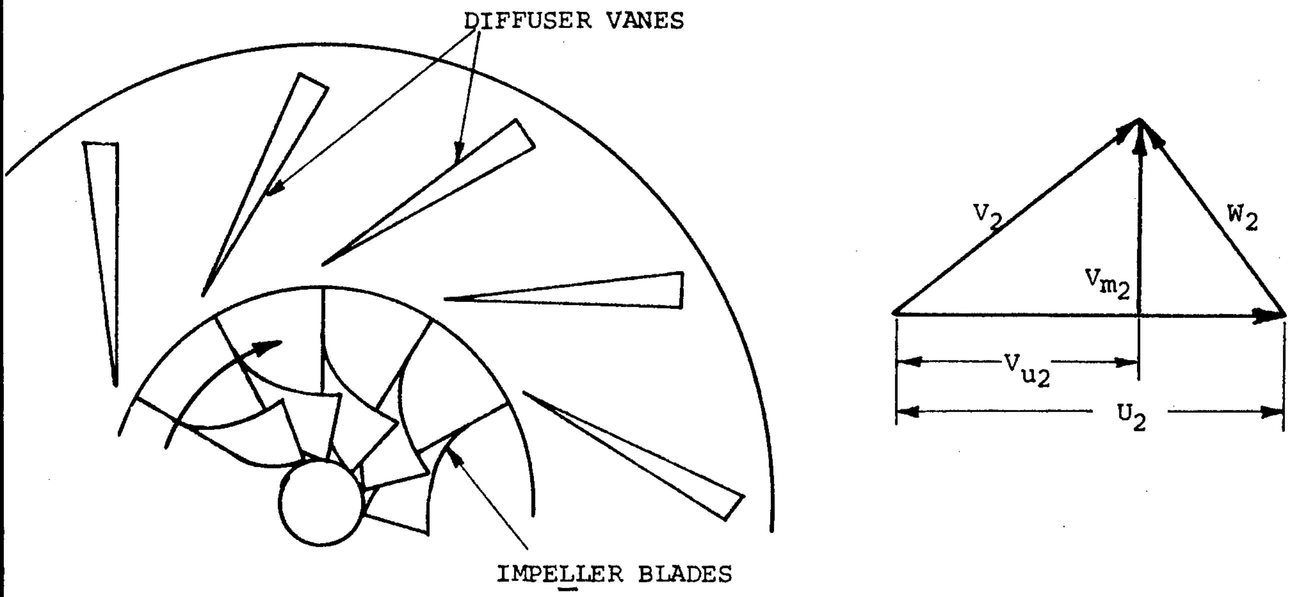
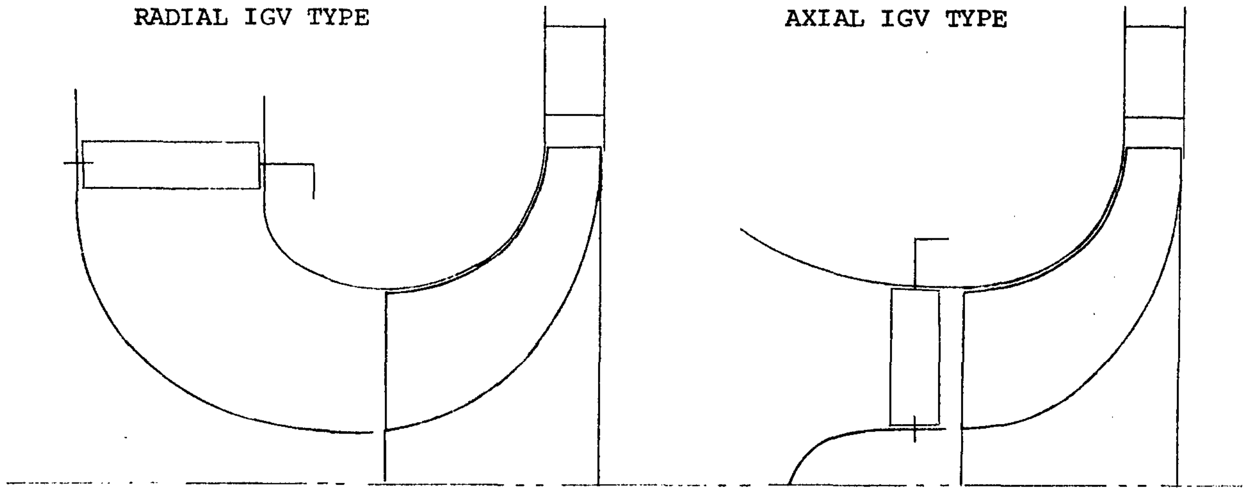


FIGURE 3.1.1(b). Impeller exit velocity diagram for a radial compressor.

inlet guide vanes, as shown in Figure 3.1.2. Provided that the inlet guide vanes are operating unstalled and there is no throttling effect (where a serious efficiency loss would be involved), the actual surge line is little changed. For the results shown in the figure there was only a small gain in flow range between the steady state gas turbine operating line and actual surge. However, the principal purpose of the variable guide vanes in these tests was not so much to change the compressor surge characteristics, but to hold rotor speed of a gas turbine power plant constant at the design value during part load conditions. Then almost instataneous power response to higher load demands could be developed by rapid actuation of the guide vanes and increase in fuel flow. Acceleration of the rotor assembly was eliminated as a consideration in load response. However, it should be noted that the success of the variable IGV system in a two-shaft gas turbine depended in part upon the characteristic of the cycle to increase gasifier turbine inlet temperature to compensate for the higher compressor power requirement at part load (as compared to a control system which allows compressor speed to change). The diesel turbocharger application differs in that turbine inlet temperature cannot be maintained in the same manner, so there is not sufficient exhaust energy to hold near-rated compressor speed while the engine idles or operates at low power output. Therefore, it is not feasible to utilize variable guide vanes in turbocharger compressors to maintain turbo speed and take advantage of the acceleration response characteristics shown for the gas turbine in Ref. 1.

Two general IGV types have been utilized. The accompanying sketch illustrates the flow path characteristics of these types of inlet guide vane approaches:



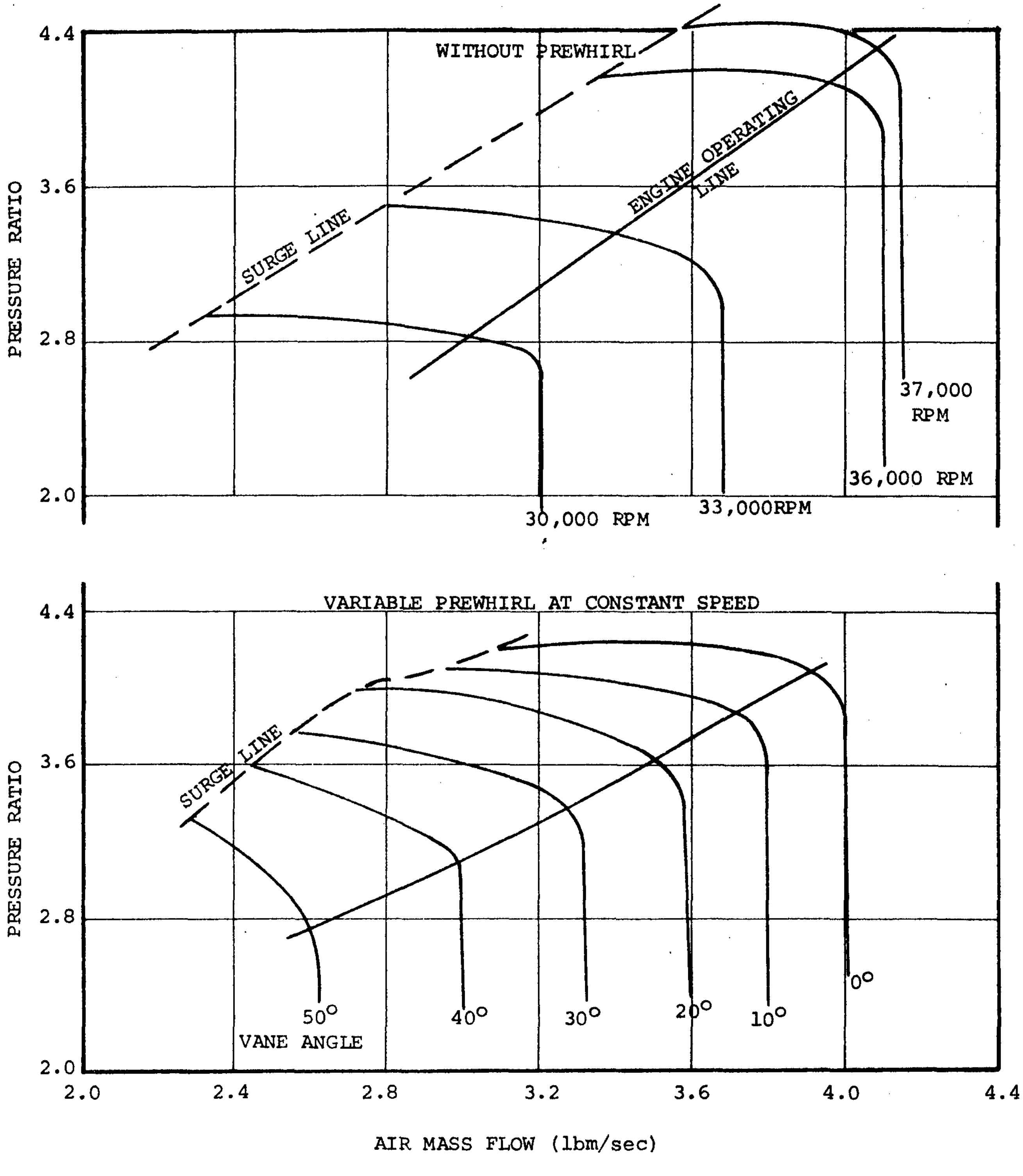


FIGURE 3.1.2. Typical Compressor Performance With and Without Prewhirl. (Ref. 1)

Historically, the radial form dates back to the original Whittle gas turbines and produces a free-vortex flow distribution at the inducer leading edge (rV_{u_1} = constant and axial velocity is constant spanwise). With a free vortex prewhirl distribution at the inlet it is very difficult to achieve a good hub to tip incidence match with the inducer unless some compromises are made with regard to good inducer structural design practices. With the axial vane type a spanwise whirl distribution more tailored to the inducer design limitations can be achieved. Tests of both the radial type and axial vanes designed to produce a constant flow angle from hub to tip were reported in Ref. 1, and the results indicated that for the same guide vane angle the pressure ratio produced by the compressor was almost identical.

One problem common to both types of inlet guide vane designs is the high incidence losses and wake losses resulting from rotation of the vanes to high stagger angles. A possible solution to this problem is the slotted/flapped airfoil, which has been tested on larger turbojets with some success in reducing guide vane losses (illustrated in Figure 3.1.3). This type of blade eliminates the stalling incidence problem because the front portion of the blade is stationary. Variable prewhirl is achieved by rotation of the flapped trailing edge to impart that amount of turning required. The slot allows a jet of high pressure air to energize the boundary layer on the airfoil suction surface, thus delaying the advent of separation to higher rates of turning. The aerodynamics of this type of airfoil are certainly superior, although a certain amount of development is required to determine the optimum slot location and dimensions (passage width, slot angle, etc.). Also, the small airfoil sizes of inlet vanes in the turbocharger size class certainly pose a more difficult manufacturing problem for the slotted/flapped type of vane. Although the losses with this type of variable vane airfoil are minimized, and this concept appears to be superior with respect to performance, unit cost considerations would seem to favor a less complicated

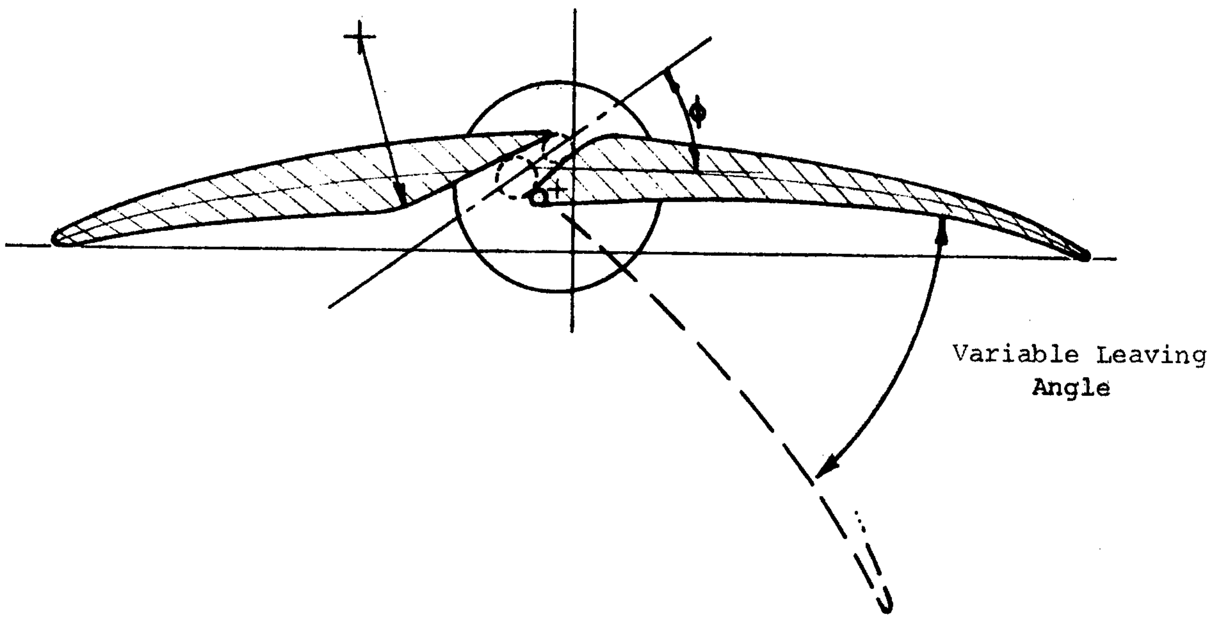


Figure 3.1.3: Slotted/Flapped Airfoil Concept for Variable Prewhirl Without Excessive Incidence Losses.

variable inlet system, particularly in light of the overall benefits to be gained from variable inlet vanes.

In summary, variable inlet guide vanes do not by themselves offer the extent of surge margin improvement or contribute to the acceleration response required of tactical military diesels. Because of the ability of variable guide vanes to keep the inlet air relative approach velocity lined up with inducer blading at part speeds, they can be used to keep the inducer operating unstalled. Variable guide vanes can be used in combination with somewhat more efficient flow reduction methods such as variable diffuser vanes to attain slightly higher efficiency at stalled impeller flow rates. However, the efficiency improvement is probably not great enough to justify the added mechanical complexity of the variable guide vane system in turbocharging applications.

3.1.2 Variable Diffusers

In the typical centrifugal compressor, the vaned diffuser component is generally conceded to be most influential in governing surge. Only at very low speeds, where the inducer tends to stall, or at speeds above design, where choking of the inducer occurs, does the impeller greatly influence surge or choke flows. It is not surprising, therefore, that regulation of IGV's has so little effect upon the surge characteristics of centrifugal compressors. Most investigators now agree that the amount of diffusion of the flow between the diffuser leading edge and the throat on the vane suction surface is most critical. When the adverse pressure gradient reaches the limiting value for that particular diffuser configuration, the machine surges. Test comparisons with various diffuser vane shapes of given throat area and vane number often show only minor changes in overall performance. Reduction of the throat area, however, reduces the diffusion between the tip and the throat and stabilizes the diffuser. With this in mind it has been suggested by Rodgers (Ref. 5) that theoretically a compressor with a variable diffuser throat downstream of an impeller essentially free of stalling and choking restrictions could provide almost constant pressure ratio down to nearly zero flow.

There are two possible means for regulating flow with the diffuser of radial compressors:

1. Reducing the width between front and rear shrouds.
2. Restagger of the vanes.

The first of these is suited mainly to the vaneless diffuser type, whereas the second, restagger of the vanes, is the most practical means of varying the diffuser throat area of vaned-type diffusers. Rotation of the vanes not only reduces the through-flow area of the diffuser but also provides an adjustment for the smaller angles of approach of the flow from the impeller at reduced flow rates.

Thermo Mechanical Systems Co. (TMSCO) analyzed available information on designs and performance of variable diffuser compressors developed by several other manufacturers, including the Solar Division of International Harvester Co., Worthington-CEI, Inc. and Escher Wyss, Ltd. of Switzerland (Refs. 5 and 6), before proceeding with a variable diffuser turbocharger design in this program. The Solar variable diffuser compressor consisted of a 5.25 in. radial impeller with sixteen blades and a diffuser consisting of fifteen variable vanes, each pivoted about the estimated design center of pressure of the airfoil. Gaps existed between the vane and shroud walls to prevent binding (.9% gap/height clearance). The Solar diffuser system is shown in Figures 3.1.4 (a) and (b) at diffuser throat areas of 100% (design) and 45%, which were two of the vane settings tested. Complete compressor performance maps generated for four different vane settings are shown in Figure 3.1.5. Airflow was found to vary nearly in relation to the diffuser throat area (i.e., surge and choke airflows at 45% diffuser throat area were about the same percentage of the design flow rates, etc.). Pressure ratio at a given speed remained relatively constant as throat area was decreased except at the very lowest throat area (16%), where efficiency decreased appreciably due to recirculation of heated, compressed air back into the inlet. The efficiency decay was 8 points at the 72% throat area and 15 points at the 45% throat area. No testing was conducted at greater than design diffuser throat areas.

Worthington-CEI, Inc. attempted to market a Modulair variable diffuser compressor in a range of airflow sizes for industrial compressor applications requiring broad flow range. This compressor, shown in Figure 3.1.6, had adjustable vaned diffusers for flow reduction, and some units, including that shown, were equipped with variable inlet guide vanes as well. The diffuser vanes in this compressor were of the tandem cascade type rather than the more conventional vane-island type tested by Solar. Another difference is that the vane airfoils were offset on the diffuser discs such that at other than design settings the leading edge of the first vane row never changes its distance from the impeller tip. An increase in the vaneless space gap occurs without use of this vane offset technique at low throat areas, and this leads to unnecessary

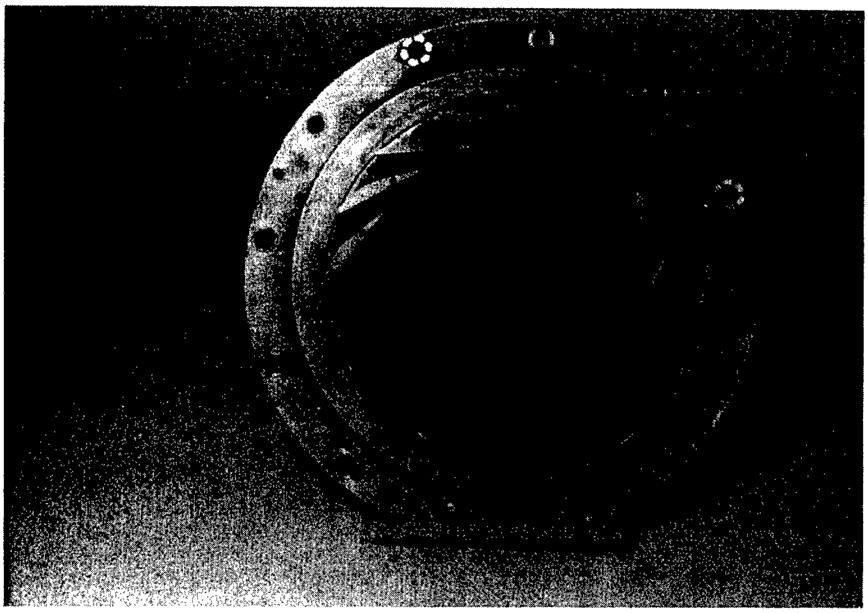


FIGURE 3.1.4(a). Solar Variable Diffuser Compressor with Vanes in Open (100%) Diffuser Throat Setting.

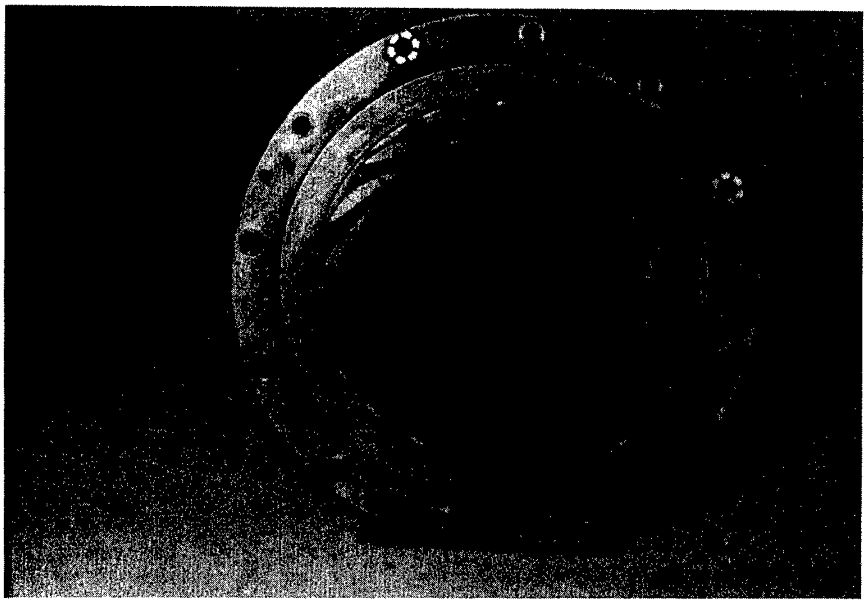


FIGURE 3.1.4(b). Solar Variable Diffuser Compressor with Vanes in Partially Closed (45%) Diffuser Throat Setting.

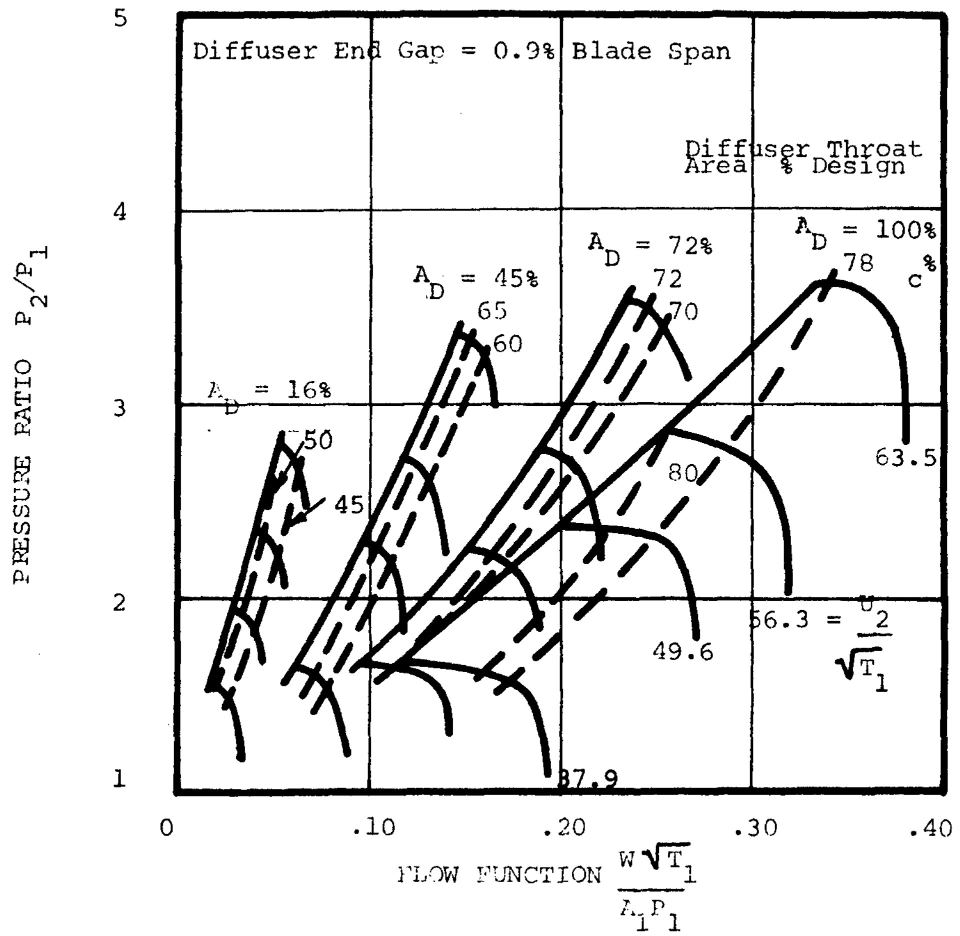
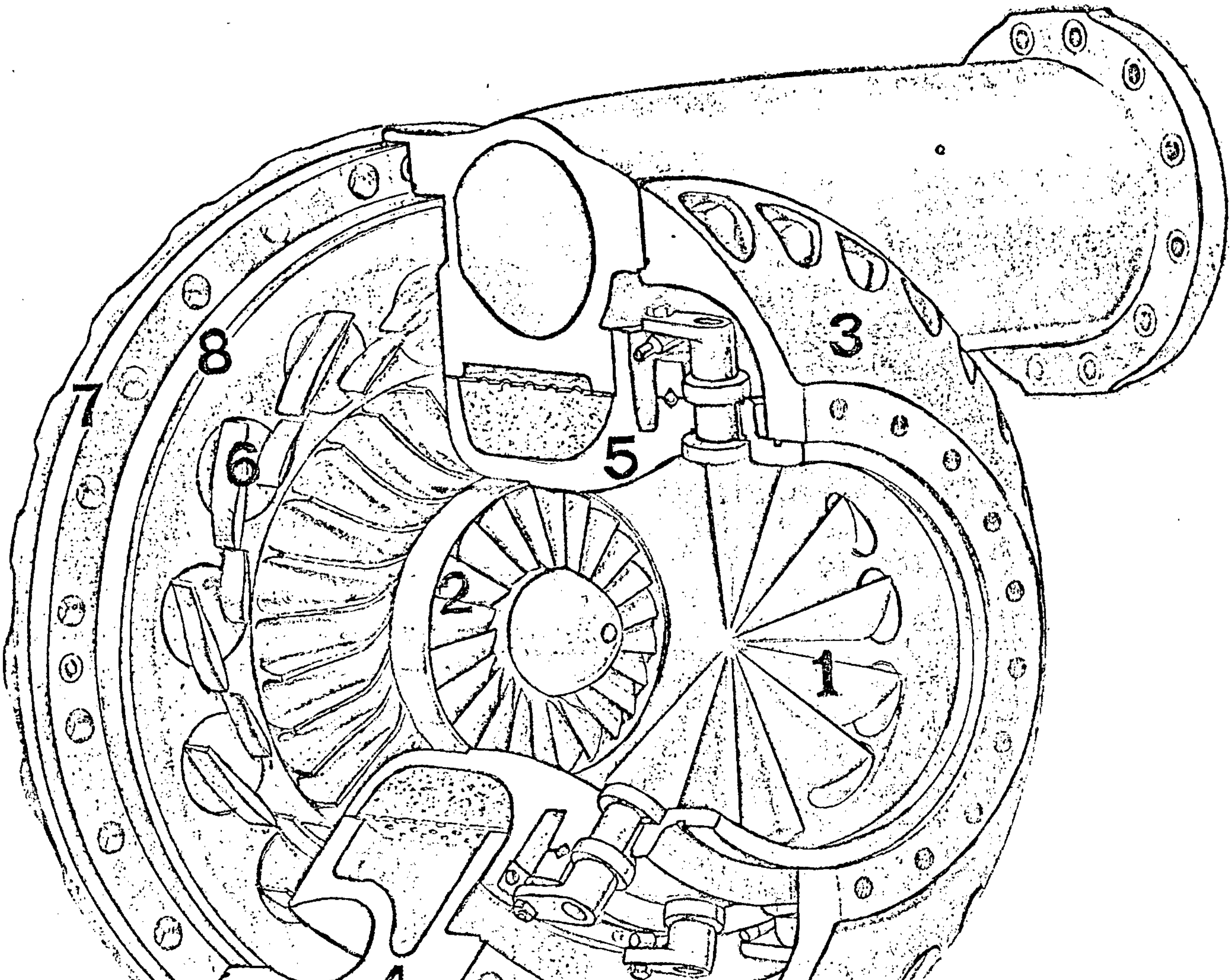


FIGURE 3.1.5. Solar Variable Diffuser Test Compressor Performance.



1. VARIABLE INLET VANES
2. IMPELLER
3. SUCTION HEAD
4. VOLUTE CASING
5. SUCTION SHROUD
6. DIFFUSER VANES
7. CASING DIAPHRAGM
8. DIFFUSER DIAPHRAGM

FIGURE 3.1.6 . Worthington Variable Geometry Radial Compressor.

friction losses and thus lower efficiency. The estimated performance of the Worthington variable diffuser compressor (without inlet guide vanes) in the turbocharger size class is shown in Figure 3.1.7. Although this compressor was never produced in sizes smaller than 10.3 in. impeller diameter, the estimated scaling effects were provided by Worthington in estimating the performance of a 5.54 in. wheel. Despite the differences in diffuser design, the general similarity in performance to the previously described Solar results can be seen. Over fifty percent reduction of flow rate was accomplished with about 15 points efficiency decay at $U/\sqrt{\theta_1} = 1032$ fps, which equals the efficiency decay in the Solar tests at 45% throat area. Pressure ratio loss at a given speed was relatively minor until approximately 50% flow reduction was obtained, where rapid pressure ratio decay and efficiency was exhibited. At the lowest vane setting, -20° of rotation of the diffuser (corresponding to about a 3:1 flow reduction), a maximum efficiency of only 39% was obtained. The Solar results indicated that 50% efficiency was obtained at 16% throat area, which suggests that the Worthington compressor experienced more severe impeller stalling and recirculation of heated air into the inlet than was observed in the Solar tests at the extreme limits of flow reduction.

Escher Wyss developed a variable diffuser system for a radial (Freon) compressor used in large refrigeration systems. This diffuser system was also of the tandem-vane type, with flow control by rotation of the first row of vanes only. This compressor had another distinct feature, stationary inlet guide vanes, to provide prewhirl and reduce the tendency of the inducer to stall at very low flow rates. The performance of this compressor is shown in Figure 3.1.8 over a range of diffuser settings and at a given speed. Over the range of flow rates tested, the decay in efficiency showed the same general trend as in the previously described tests; that is, at lower flows reduced head and efficiency was obtained. A

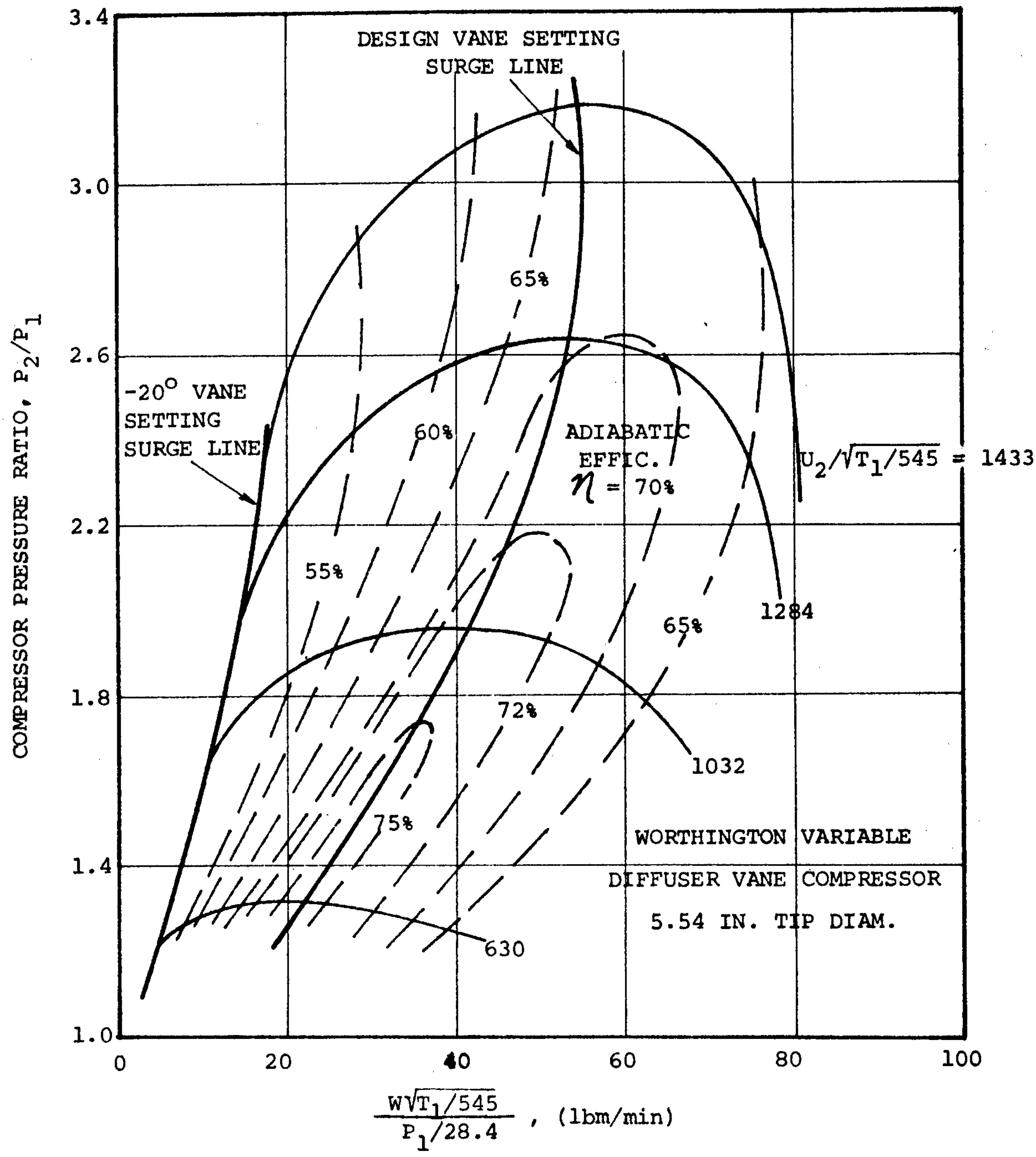


FIGURE 3.1.7. Estimated Performance of Worthington Modular Variable Diffuser Compressor in the Turbocharger Size Class.

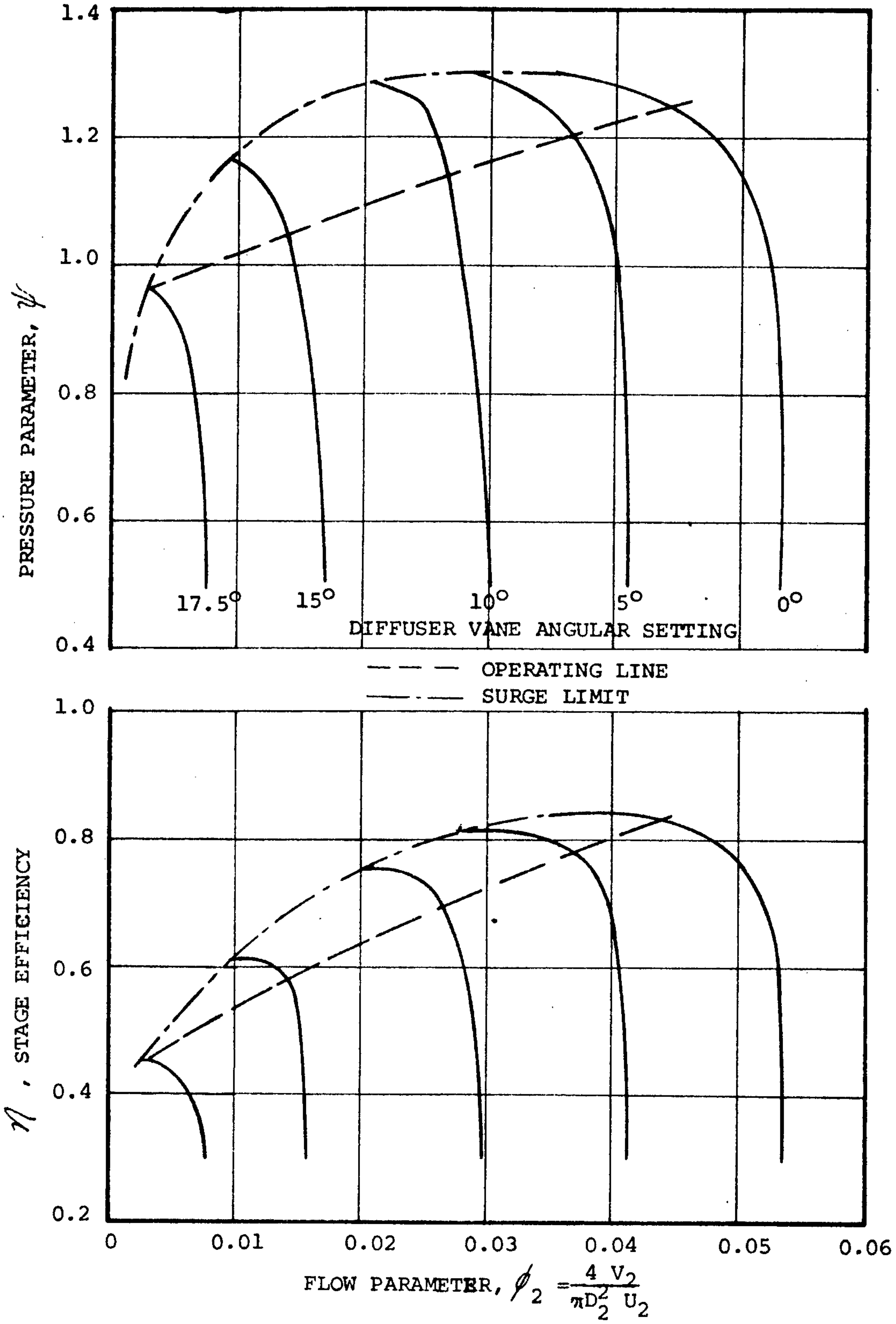


FIGURE 3.1.8. Performance of Escher Wyss Variable Diffuser Compressor.

comparison of efficiency with the Solar results is illustrated in Figure 3.1.9 . The peak efficiency at reduced diffuser throat settings is plotted for both compressors, showing the trend of efficiency loss with reduced flow rate. At comparable flow rates the efficiency decay in the Escher Wyss tests was less (perhaps 3 points). Although these compressors were designed for different pressure ratios and different working fluids, comparison of the non-dimensional head coefficient (ψ) of the two designs showed close similarity (1.25 for the Escher Wyss design, 1.26 for the Solar design). Therefore, the slightly more favorable performance of the Escher Wyss compressor was judged to be due to:

1. The entire system (inlet, impeller, diffuser) was specifically designed for variable diffuser geometry, whereas in the Solar system the impeller geometry was not.
2. The inlet guide vanes and high solidity impeller blading reduced the loading and improved the stability of the Escher Wyss impeller at reduced flow rates.

Optimum Performance with Variable Diffuser Geometry

In reporting on the test results obtained with the Solar variable diffuser system, C. Rodgers of Solar (Ref. 5) reported that the impeller geometry tested had not been specifically designed to operate in a variable geometry system. A blading optimization technique developed at Solar was used to examine alternate configurations of the compressor which might improve the efficiency of the variable diffuser compressor at partial flow conditions. The effect of flow reduction on efficiency for several configurations is shown in Figure 3.1.10. Of the configurations examined, the minimum efficiency decay was obtained with:

1. Relatively large impeller diameter ratio, $\epsilon = 2.5$ (defined as tip diameter/inducer RMS diameter), compared to the tested geometry ($\epsilon = 1.96$).
2. Prewhirl of 30° provided by inlet guide vanes.

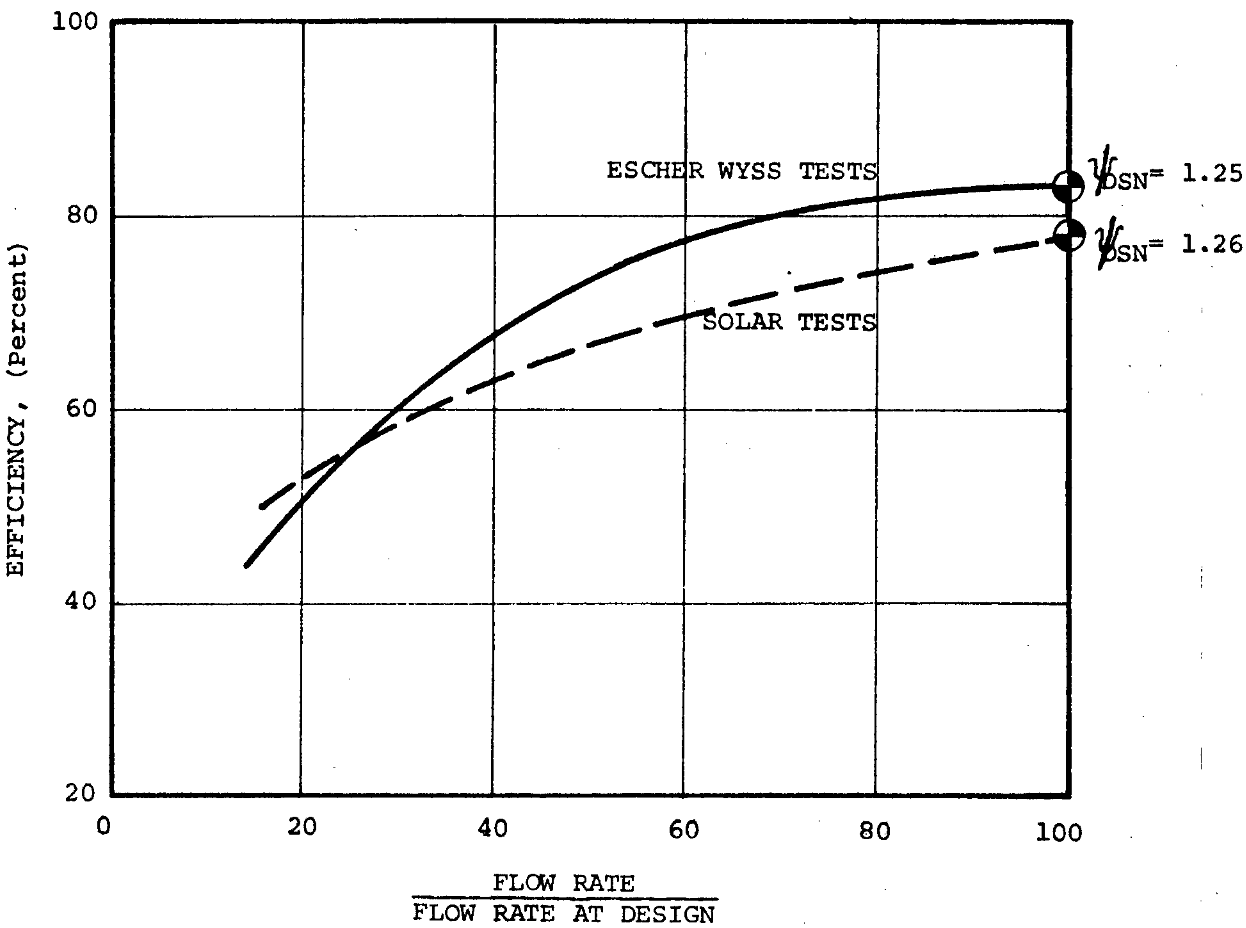


FIGURE 3.1.9. Comparison of Efficiency Decay of Escher Wyss and Solar Compressor with Actuation of Diffuser Vanes at a Given Speed(Design).

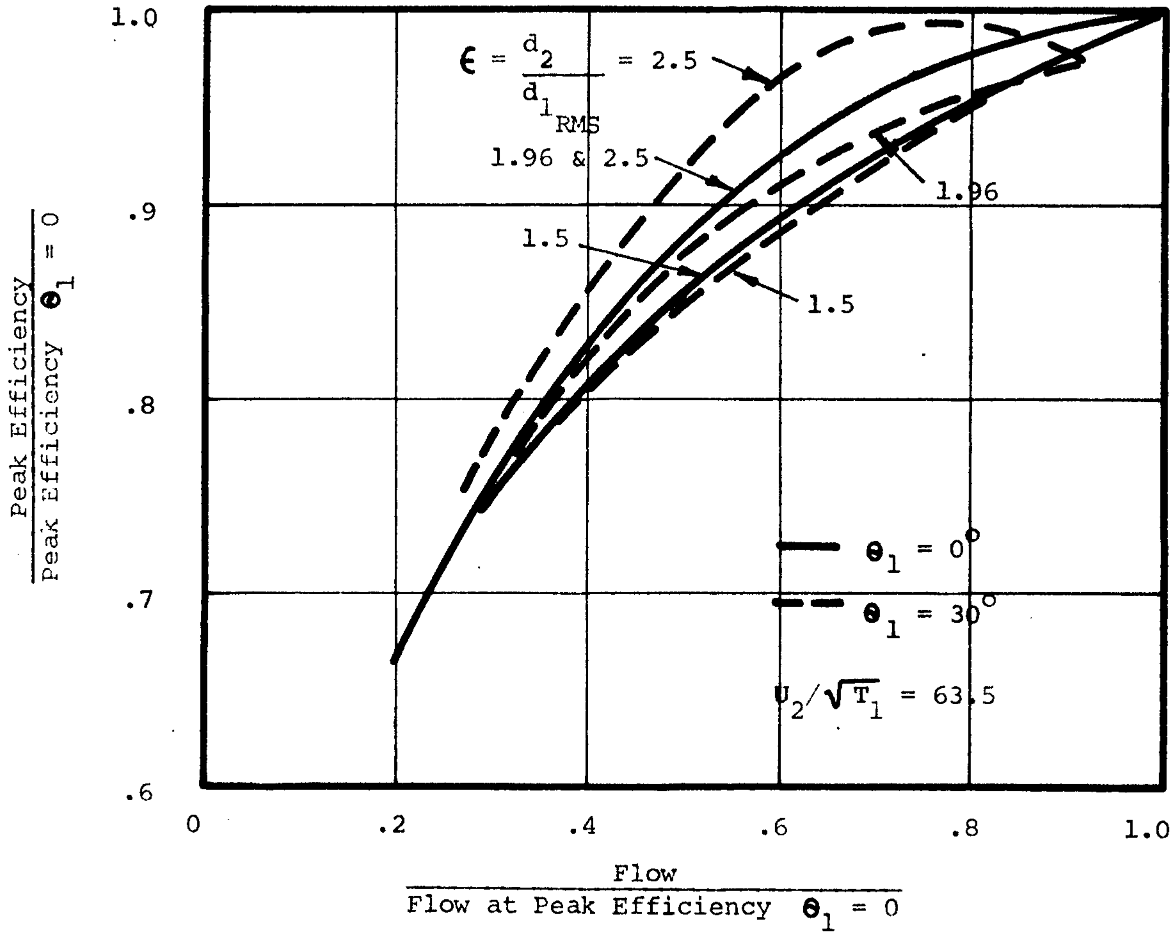


FIGURE 3.1.10. Estimated Effect of Variable Diffuser Area on Efficiency for Selected Geometric Configurations Computed at a Given Speed **(Ref. 5)**.

This analysis indicated that with optimum impeller geometry selection, some degree of prewhirl, and a variable diffuser, the flow can be reduced by 40 percent with a loss of only about 4% in efficiency. The combination of large diameter ratios and inlet prewhirl permit the inducer to experience the minimum change of incidence as flow rate is reduced. Therefore, anything which can be done to lend stability to the inducer without compromising the peak efficiency of the compressor should be considered. However, prewhirl is an added complication to the system which is probably not justified by the performance benefits to be gained. From Figure 3.1.10 it can be seen that the difference in efficiency between the best prewhirl configuration and the optimized zero prewhirl case ($\theta_1 = 0^0$) is only 2 to 4% efficiency improvement over a very broad range of flow rates.

Other means of increasing stability at off design conditions include:

1. High impeller blade number (increased solidity) for reduced aerodynamic loading.
2. Minimum number of diffuser vanes to reduce the possibility of a separated wake from the impeller filling the diffuser channel.
3. Optimized diffuser vane incidence with flow reduction by selecting the proper location of the vane on the pivot shaft.
4. Proper inlet design to insure a stable smooth flow distribution at impeller face.
5. Optimized compressor size to insure that the compressor can handle the full range of flow rates required at the highest possible level of efficiency.

With regard to the last optimization technique, it is usually possible to utilize somewhat higher than design diffuser throat areas to increase flow rate beyond the normal "design" airflow before substantial loss of efficiency due to choking is evidenced. This results because fixed geometry compressors are often compromised somewhat in peak efficiency to obtain maximum flow range; that is, a smaller diffuser throat area than that which produces maximum efficiency is selected (low ratios of diffuser throat to compressor inlet area produce maximum surge range). Therefore, by opening the diffuser throat area it is possible to substantially increase the flow swallowing capacity of the compressor until the inducer choked flow limits are reached. Then a smaller compressor size can be selected for a given maximum flow requirement, which has several important benefits:

1. The smaller compressor can handle lower flow rates at a higher percentage of its "design" diffuser setting, thereby minimizing the efficiency decay inherent in large throat area adjustments.
2. The smaller compressor is likely to have lower inertia (thus maximum acceleration capability).

There are many interrelated factors involved in an optimum variable geometry selection and for this reason the optimization techniques previously mentioned must be carefully applied. Some characteristics of the system improve the system at one end of the operating spectrum but hinder at the other. For example, high diameter ratio (ϵ) was previously shown beneficial in stabilizing the inducer and reducing the efficiency decay at lower diffuser settings ($A_D < 100\%$). However, at high compressor speeds (high pressure ratio) and higher than design diffuser throat settings, it is found that lower diameter ratios (1.5) give much higher efficiency (Figure 3.1.11). Geometry must be carefully selected to insure that a proper match between performance at high and low flow conditions is provided.

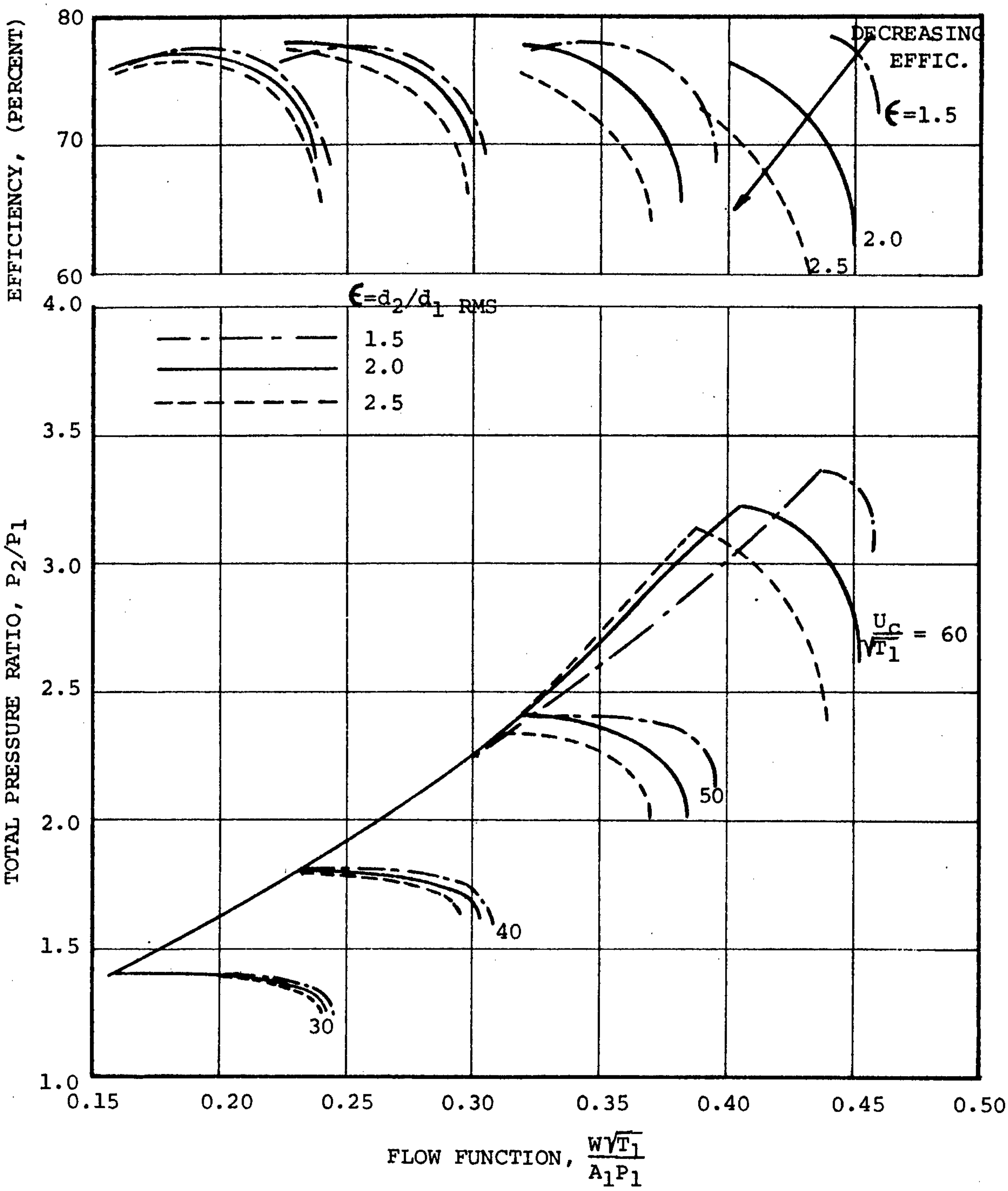


FIGURE 3.1.11. Comparison of Compressor Performance at Different Diameter Ratios (ϵ) for a Compressor with Greater than Design Diffuser Throat Area (Ref. 8).

3.1.3 Aerodynamic Methods of Surge Control

The attraction of aerodynamic methods of surge control is that relatively complex variable geometry mechanisms could be dispensed with if flow regulation could be efficiently accomplished by simpler means. Three basic forms of controls have been used on centrifugal compressors:

1. Jet re-entry or recirculation of compressor discharge air.
2. Blowoff of compressor discharge air.
3. Boundary layer control (bleed).

Laskin and Kofskey of NACA described tests (Ref. 9) of a "surge inhibitor" which utilized a recirculated jet of high energy compressor discharge air ducted back to the inlet to aerodynamically impart varying amounts prewhirl in the direction of rotation of the wheel. The results of these tests are illustrated in Figure 3.1.12. Considerable displacement of the surge line was observed, especially for the highest operating speed where the maximum flow was limited by impeller choking. Only under these conditions, however, is simulated IGV regulation particularly effective. In addition, all surge margin gains were obtained at the expense of heavy loss of pressure ratio and efficiency. At lower speeds the surge line displacement is less, but efficiency and pressure losses are still evident. These losses are due to unavoidable streamwise mixing losses, the effect of prewhirl on compressor work, and higher compressor inlet temperature. Most importantly, in all cases the compressor must perform work on the net airflow plus the recirculated airflow, and usually at extremely low levels of efficiency. The compressor horsepower increase is directly proportional to the airflow and inversely proportional to efficiency. With 40% recirculation the actual flow through the compressor is about 40% more than the gas flow available at the exhaust turbine, so it appears to be a near impossible task to supply the necessary compressor power (even without considering efficiency) at reasonable diesel engine back pressures. This method of surge control has been deemed impractical for this application.

Another method of surge control proposed for diesel turbocharger

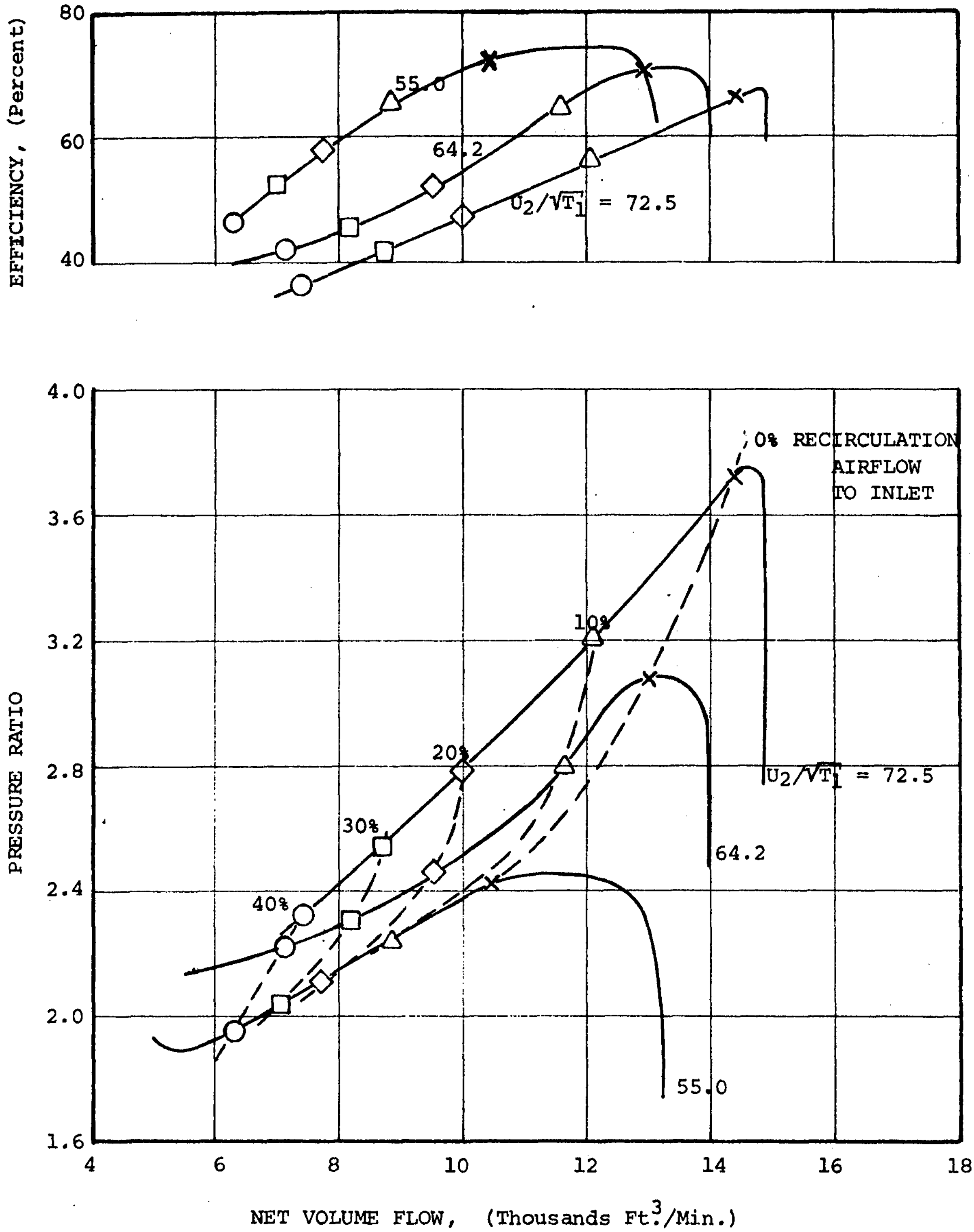


FIGURE 3.1.12. Effect of Inlet Jet Re-Entry on Flow Range and Efficiency of a Centrifugal Compressor.

compressors involves the use of a blowoff valve on the compressor discharge line. In this system, the compressor continues to flow an amount larger than that at which it surges, but the reduced airflow requirement of the engine is met by dumping excess compressed air overboard. A similar technique has been used successfully in aircraft turbojet engines, and even though wasteful this is felt to be a much more reasonable approach than the jet re-entry technique. Figure 3.1.13 illustrates the surge line comparison of the compressor from the jet re-entry tests at 40% recirculated airflow compared to the same compressor with 40% blowoff. The advantages of the blowoff approach include greater surge margin (lower flows are possible at a given pressure ratio) and the compressor horsepower requirement is reduced by about 1/3 due to higher efficiency compared to the inlet jet re-entry system, where efficiency was so drastically affected. Its principal shortcoming is that for large increases in surge margin an unacceptable compressor horsepower requirement is still involved relative to the power which can be derived from a turbine operating at perhaps 50% less flow rate. Therefore, this surge control technique appears useful only if minor improvement of compressor operating range is required and an effective boost augmentation system can be provided.

A third technique of surge control which can be utilized is boundary layer control. With regard to surge, the most critical area in the centrifugal compressor is along the suction surface of the diffuser vane between the tip and the throat. As the surge flow rate is approached the boundary layer along this short section of the vane becomes unstable and finally separates, triggering surge. By incorporating small bleed holes or slots along this surface it should be possible to remove or transfer the unstable boundary layer and avert surge. Bleed slots have been incorporated in test hardware at Solar (Ref. 4) and have proved successful to a limited degree in reducing surge flow rates. Results with relatively large amounts of bleed suction (15 to 20 percent) showed an improvement in flow range at the expense of an efficiency loss. However, the level of operating range improvement desired for advanced

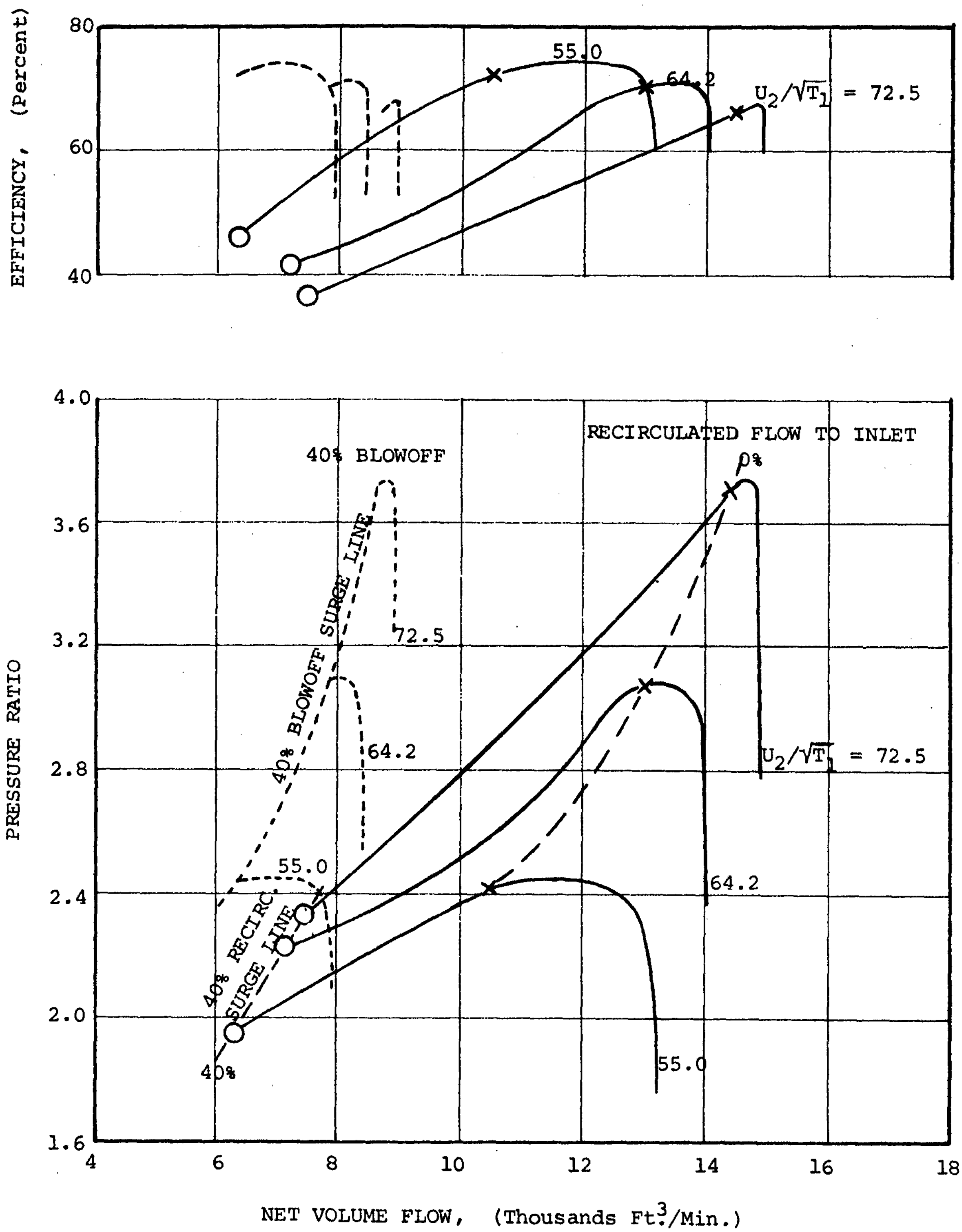


FIGURE 3.1.13. Comparison of Surge and Efficiency Characteristics of Centrifugal Compressor with (a) Surge Control by Jet Re-Entry and (b) Surge Blowoff Valve.

diesel turbocharging, i.e., unlimited boost at any engine operating speed, requires more control of surge than any of the known boundary layer bleed control schemes can realistically provide. The practicality of a bleed system is further hindered by the relatively small size of turbocharger hardware.

In summary, none of the known aerodynamic methods for surge control appear to be able to challenge the variable diffuser system for regulating airflow of centrifugal compressors for diesel turbocharging. Recirculation and blowoff methods of surge control are wasteful of available exhaust turbine power compared to variable diffuser compressors, and boundary layer control techniques lack the ability of achieving a sufficient flow reduction.

3.1.4 Special Configurations of Rotor Blades and Diffuser Vanes for Surge Control

Current high pressure ratio diesel turbocharger compressors conventionally utilize radial impeller blades for ease of casting and minimum stresses, enabling the entire compressor assembly to be inexpensively cast from a lightweight aluminum alloy. Vaned diffusers are commonly chosen to achieve reasonable efficiency, although the overall design is often compromised somewhat in peak efficiency to provide adequate operating range for the particular application. The conventional turbocharger compressor is not particularly optimum from any performance standpoint, but does satisfy current diesel performance criteria in an acceptable manner as well as provide low cost and high reliability. Other configurations can be proposed which offer potential performance improvement for advanced diesel applications, most of which increase the complexity and cost of the compressor component. The following compressor concepts will be discussed in the following section:

1. Backswept compressor.
2. Multi-stage compressors.
3. Vaneless diffusers, including the free-rotating diffuser concept.

Backswept Compressors

It is generally known that impeller tip backsweep affords higher efficiency and broader surge range potential than is possible with radial bladed compressors. The backswept compressor performance advantages stem from a lower inlet absolute Mach number into the critical diffuser component and reduced impeller aerodynamic loading. Until recently, however, radial blades were used for all but the very low pressure ratio, low tip speed applications because blade bending stresses increase with backswept blading. With more sophisticated stress analysis techniques and newer materials such as titanium now available, it is possible to use impellers with backsweep up to tip speeds of about 2000 fps. At a pressure ratio of 4-5:1 the potential efficiency benefit appears to be between 3 and 5 percent. The flow range advantage of backswept rotors is illustrated by Figure 3.1.14 from Ref. 4. At a given diffuser inlet Mach number the diffuser achieves a slightly lower surge/choke flow ratio with a backswept rotor. However, the design diffuser inlet Mach number is also lower for the backswept compressor, so the range advantage is magnified.

A backswept rotor design was analyzed for the VHO turbocharger compressor application. Figure 3.1.15 illustrates the broad range and high efficiency (85% peak) of this 50° backswept design. Compared to the standard compressor surge line (shown dashed) there is a potential surge margin advantage at lower speeds where diesel boost is severely limited, although not enough to provide the ultimate goal of unlimited boost at any engine speed. Ten to twelve points efficiency advantage is potentially available over the entire diesel operating spectrum. In the conventional turbocharged diesel system it has been found that compressor efficiency has a minor effect upon overall engine performance, but in the turbocompound cycle being studied for advanced diesels an efficiency increase of this magnitude assumes greater importance.

Backswept compressors combined with variable diffuser geometry offer the greatest potential for high efficiency at reduced flow rates. If less flow reduction capability is required than that afforded by variable diffuser geometry, it should be possible to control surge with a simple blowoff valve in the compressor discharge line without causing an excessive compressor power requirement. The inherent stability of the backswept rotor reduces the amount of blowoff necessary to attain a given surge margin and higher efficiency partially compensates for turbine power wasted in compressing excess air.

The apparent performance advantages of the backswept compressor must be carefully weighed against the certain structural advantages of the conventional cast radial wheel. Higher blade stresses caused by swept blading and higher wheel speeds (necessary to match the pressure rise of a radial compressor of comparable efficiency) require a stronger, heavier

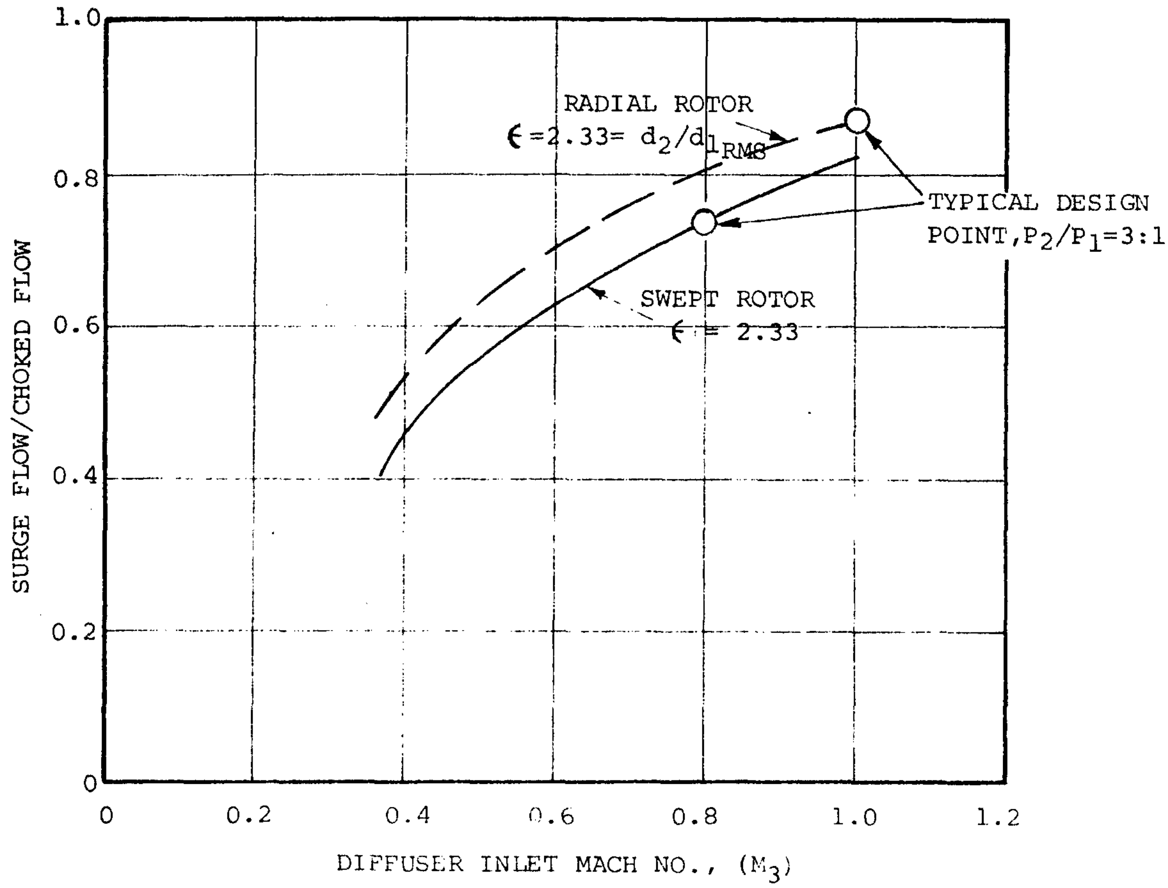


FIGURE 3.1.14. Comparison of Flow Range of Centrifugal Compressor Diffusers (Ref. 4).

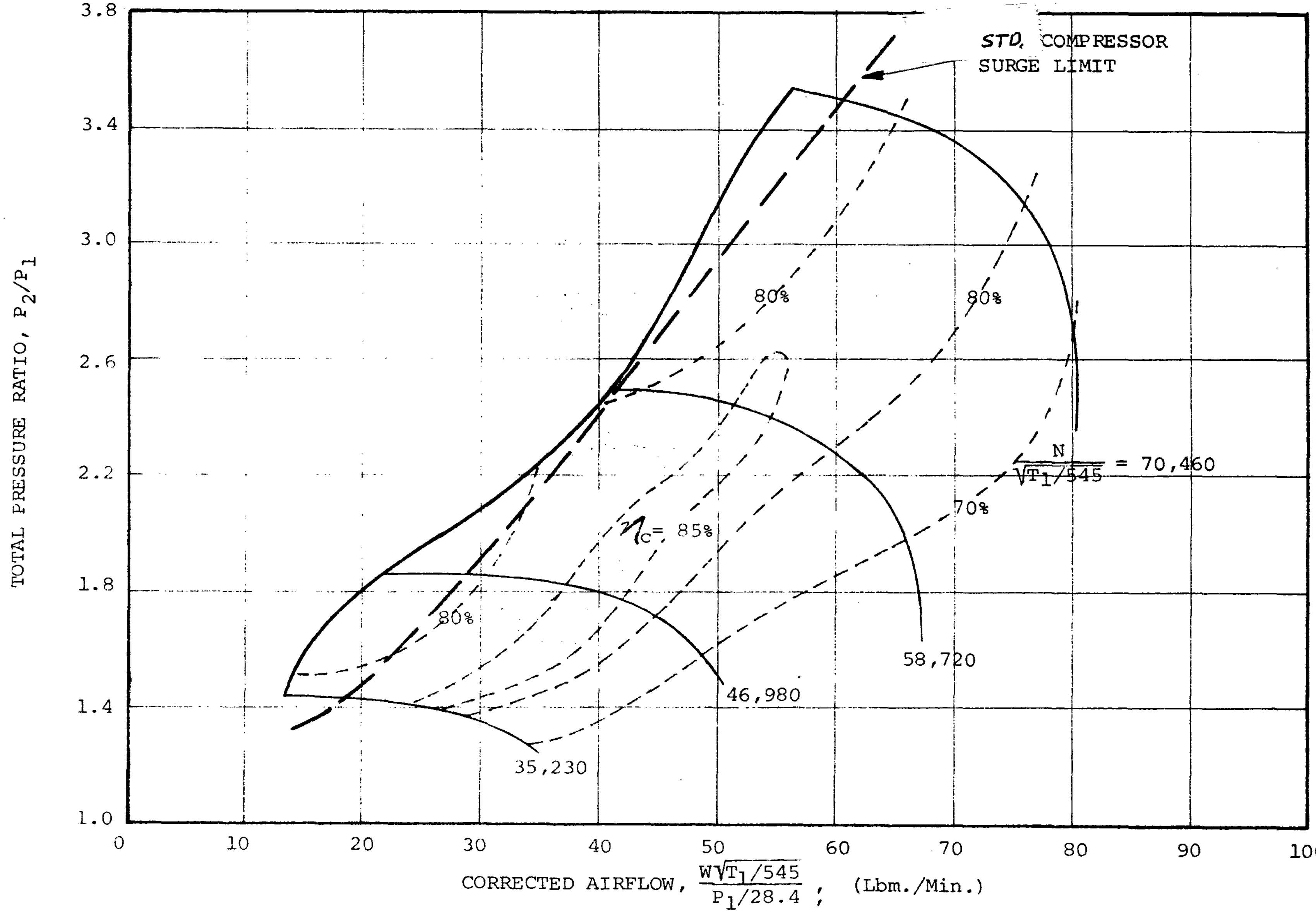


FIGURE 3.1.15. Backswept Compressor Design Analyzed for VHO Diesel Turbocharging Application.

material like titanium or steel. Inertia of the turbocharger is increased by direct proportion to weight, so acceleration of the turbo suffers. Increased stresses also limit the pressure rise capability to about 5:1 with 30 to 40 degrees of backsweeep, and lower amounts of backsweep reduce the performance advantage. But for future high performance diesel turbocharging applications where cost is secondary and performance is primary, the backswept compressor should be considered.

Multi-Stage Compressors

As diffuser inlet absolute Mach numbers increase, the attainable range of centrifugal compressors, especially the radial bladed version, tends to decrease. It is possible, however, to reduce the inlet velocity to the diffuser component by splitting the required pressure ratio into two or more stages. Often this is accomplished in gas turbine compressors by combining one or more axial stages upstream of a centrifugal stage. This gives a reasonable compromise between flow range, efficiency, and a workable flowpath. For the turbocharger application, where maximum flow range is a more critical requirement than high efficiency, a dual centrifugal stage could be proposed. For compressor pressure ratios at or above 5:1, where single stage backswept compressors reach tip speed limitations and radial compressors exhibit rather short flow range, a two-stage backswept compressor is a reasonable fixed-geometry alternative. This configuration cannot match the flow range of variable diffuser compressors, but for less severe flow range requirements the two stage centrifugal compressor can be considered as a reasonable alternative to the mechanical complication of variable geometry actuation systems. An important disadvantage is the increased rotational inertia of the compressor component, which is a primary factor in the response of the turbocharger to changes in load and speed of the engine.

Vaneless Diffusers

Vaneless diffusers characteristically attain better flow range than any of the vaned diffuser configurations (except variable geometry versions). This occurs because in the diffuser the major diffusion occurs along a relatively long spirial flow path rather than in the short distance from the tip of the vane to the throat, and in addition there is an absence of impeller-diffuser matching sensitivity. However, the diffuser friction losses increase rapidly as vaneless diffusion above pressure ratios of about 2.5:1 is attempted, so current turbocharger compressors commonly utilize a combination of relatively wide vaneless space region and vanes to achieve the best compromise between surge range and efficiency. Another fixed geometry alternative which would appear to have merit for compressor pressure ratios between 4 and 5:1 is the two-stage backswept compressor with vaneless diffusers. By splitting the pressure ratio the diffuser entry Mach number is reduced to a level where reasonable efficiency could be obtained. The performance of a two-stage highly backswept compressor (Ref. 10) is shown in Figure 3.1.16, with airflow scaled to the VHO engine requirement. Although the vaneless type diffuser was used only on the second stage, considerable improvement in surge margin is shown relative to the standard radial compressor. It is concluded that a configuration combining two backswept compressor stages with vaneless diffusers promises the greatest potential surge range of any of the fixed geometry alternatives yet discussed. The disadvantages are that its high inertia reduces response time and that the conventional simple and compact turbocharger mechanical design cannot be utilized with the larger overhung mass on the two-stage compressor.

Another concept which is getting some recent attention is the free-rotating vaneless diffuser. Solar has published the results from an initial study of a free-rotating vaneless space diffuser designed and developed under a U.S. Army Mobility Research and Development Contract (Ref. 11). A sketch of the concept is shown in Figure 3.1.17. In this system, the diffuser vaneless space walls and vanes are allowed to rotate independently of the rotor such that the diffuser rotates at a major

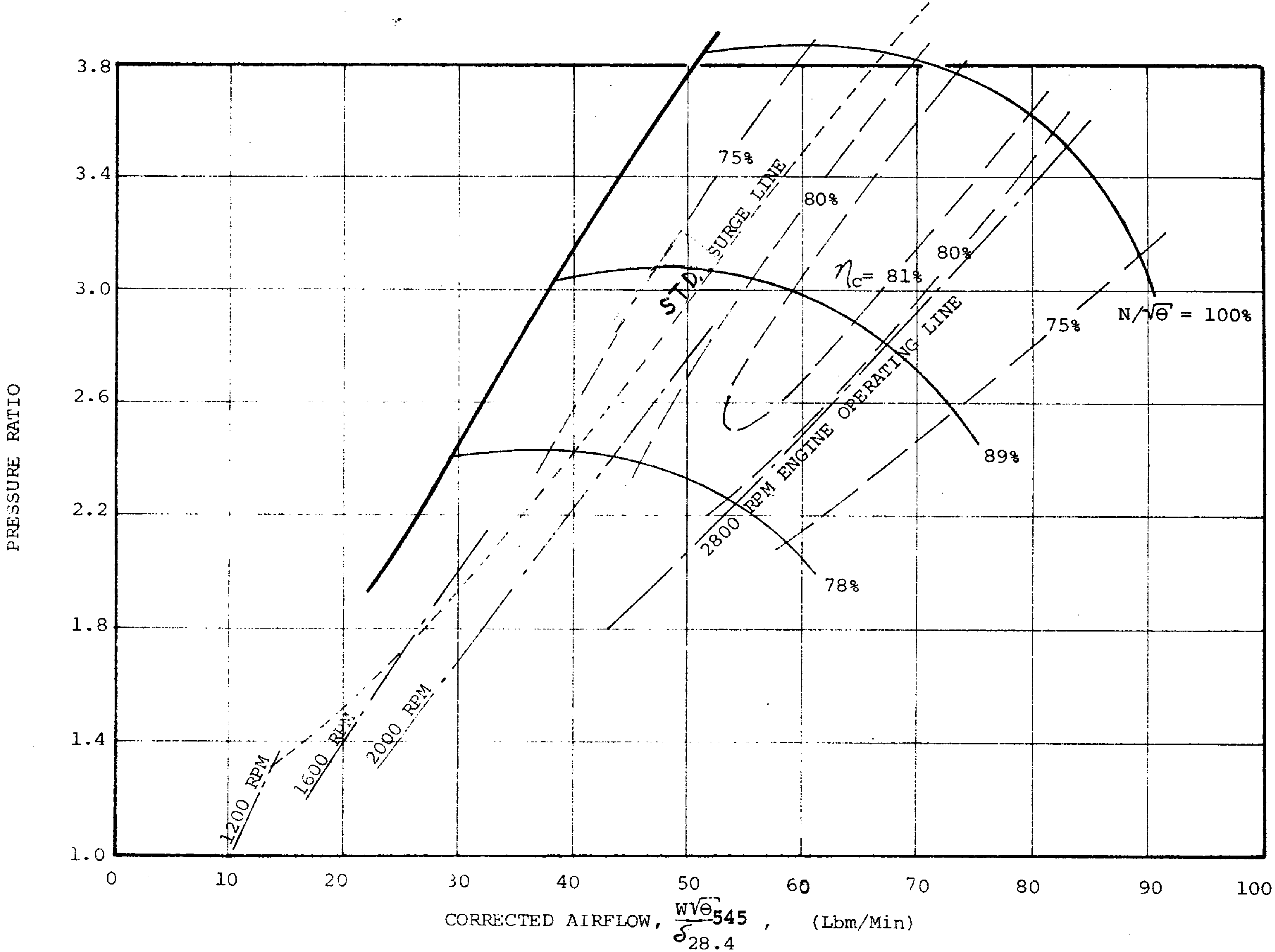


FIGURE 3.1.16. Comparison of Surge Range Available with Two-Stage Backswept Compressor Relative to Standard Radial Compressor used on VHO (Ref. 10).

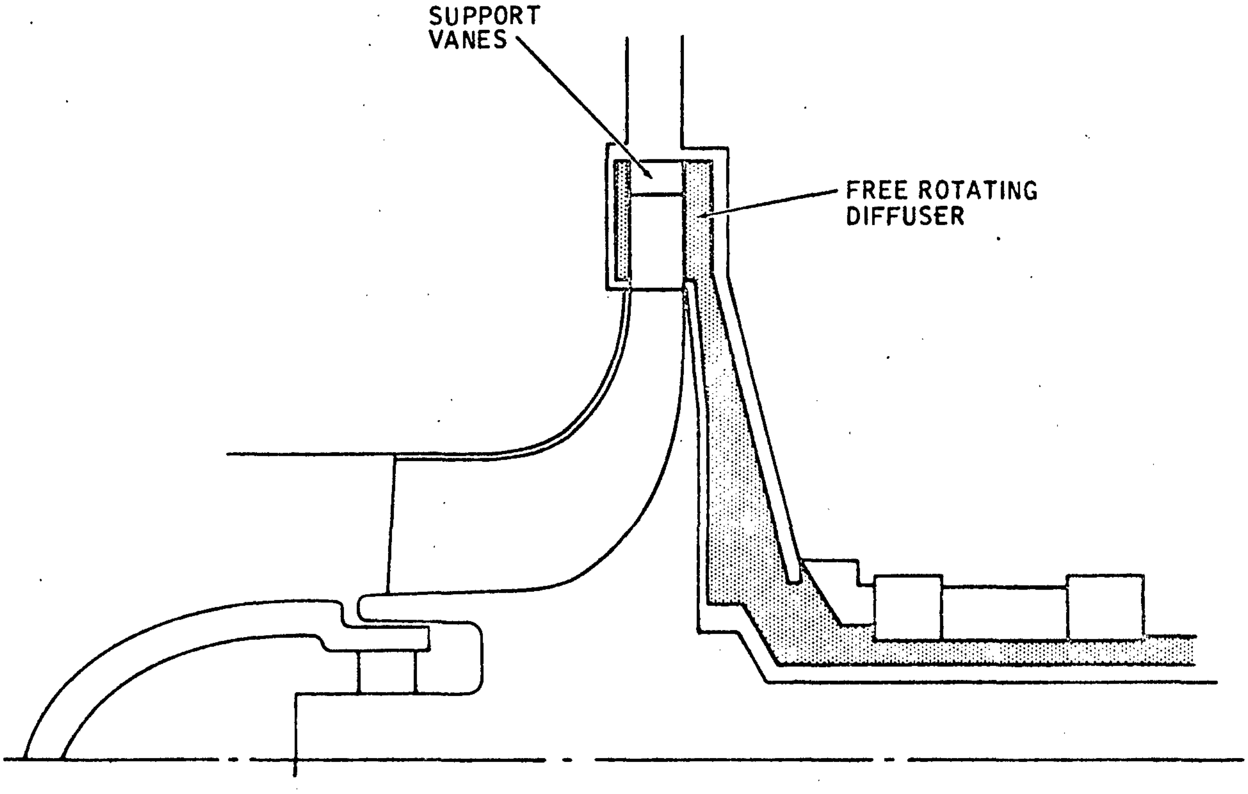


FIGURE 3.1.17: Basic Arrangement of SOLAR Free Rotating Diffuser.

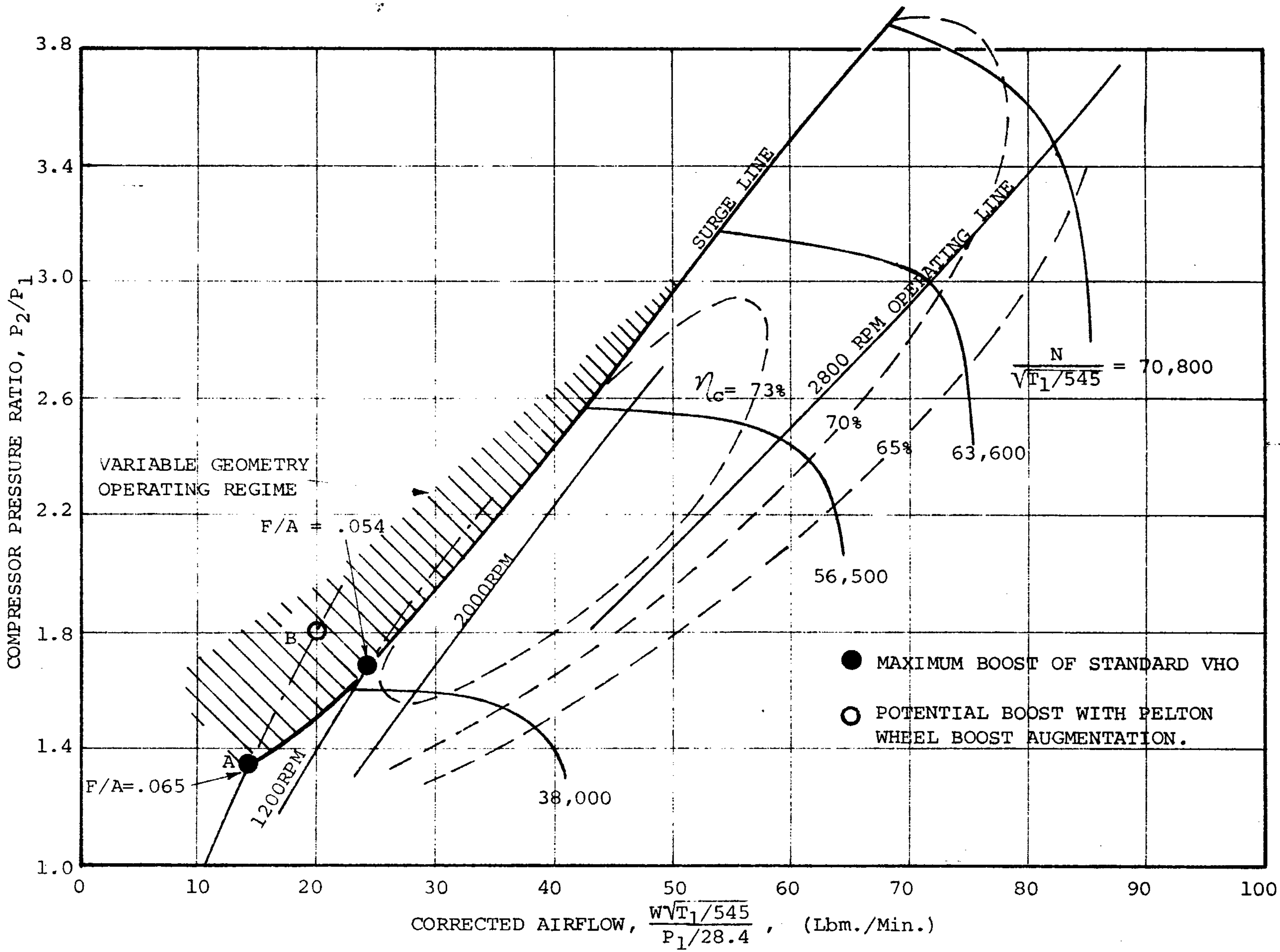
fraction of the rotor speed. Then the shear forces between the high velocity flow leaving the impeller and the walls are greatly reduced. Efficient diffusion of the high impeller exit absolute Mach numbers is accomplished in the relatively long vaneless space with only about 20% of the attendant friction losses in a stationery wall diffuser of the same length. The rotation of the sidewalls prevents separation, tends to smooth the distorted flow profiles from the impeller, and provides stability for the vaned portion because of lower velocities entering the vaned region. Since the diffuser controls surge over the normal compressor operating range, a broader range compressor map results. When used in combination with an advanced backswept impeller, this concept might provide a degree of surge range for high pressure ratio compressors thought previously to be attainable only with variable geometry. Unfortunately, published performance of an impeller diffuser combination is not yet available; previous results were attained with simple diffuser flow tests.

3.2 Analysis of Potential of Pelton Wheel and Other Boost Augmentation Systems

Conventional diesel turbochargers are limited to relatively low boost levels at low and medium engine speeds by not only the compressor surge problem but also the available engine exhaust energy. Figure 3.2.1 illustrates the VHO compressor map with engine operating lines superimposed. Operation in the shaded region is possible at 1200 RPM with some effective means of surge control (i.e., variable diffuser geometry) only if enough power can be generated in the exhaust turbine to drive the compressor to higher pressure ratios. At point A, however, the VHO is operating at a near stoichiometric fuel/air ratio (.065), so dumping additional fuel into the cylinder will not prove effective in increasing the exhaust temperature or energy. In addition, the smoke level at this engine fuel/air ratio is high. A similar situation exists at 1600 RPM.

The possibility of utilizing a Pelton impulse turbine wheel supplement to the turbine drive of a diesel turbocharger was suggested by Timoney (Ref. 12) and is shown schematically in Figure 3.2.2. A high pressure hydraulic pump is driven from the engine crankshaft and supplies high pressure (engine) oil to the single hydraulic nozzle. There it is converted to a high velocity jet which impinges on the turbine buckets and transfers energy to the turbo shaft. It is intended that such a hydraulic turbine would be used primarily to provide power to the compressor during startup, to increase boost under conditions of low exhaust gas energy, and as an aid in accelerating the turbocharger rapidly from low to high speeds, thereby improving the response of the diesel engine.

If the Pelton wheel system is incorporated into the turbocharging of the VHO engine, it is possible (for example) to use engine power to drive the compressor at the 35% higher boost level of point B on Figure 3.2.1. The total compressor horsepower required to operate at this point is 18.7 HP, part of which will be supplied by the exhaust turbine and the rest coming from the Pelton turbine drive. If the boost pressure



CORRECTED AIRFLOW, $\frac{W\sqrt{T_1/545}}{P_1/28.4}$, (Lbm./Min.)

FIGURE 3.2.1 Compressor Map Showing Higher Boost Possible at Low Speeds with Pelton Drive Supplement.

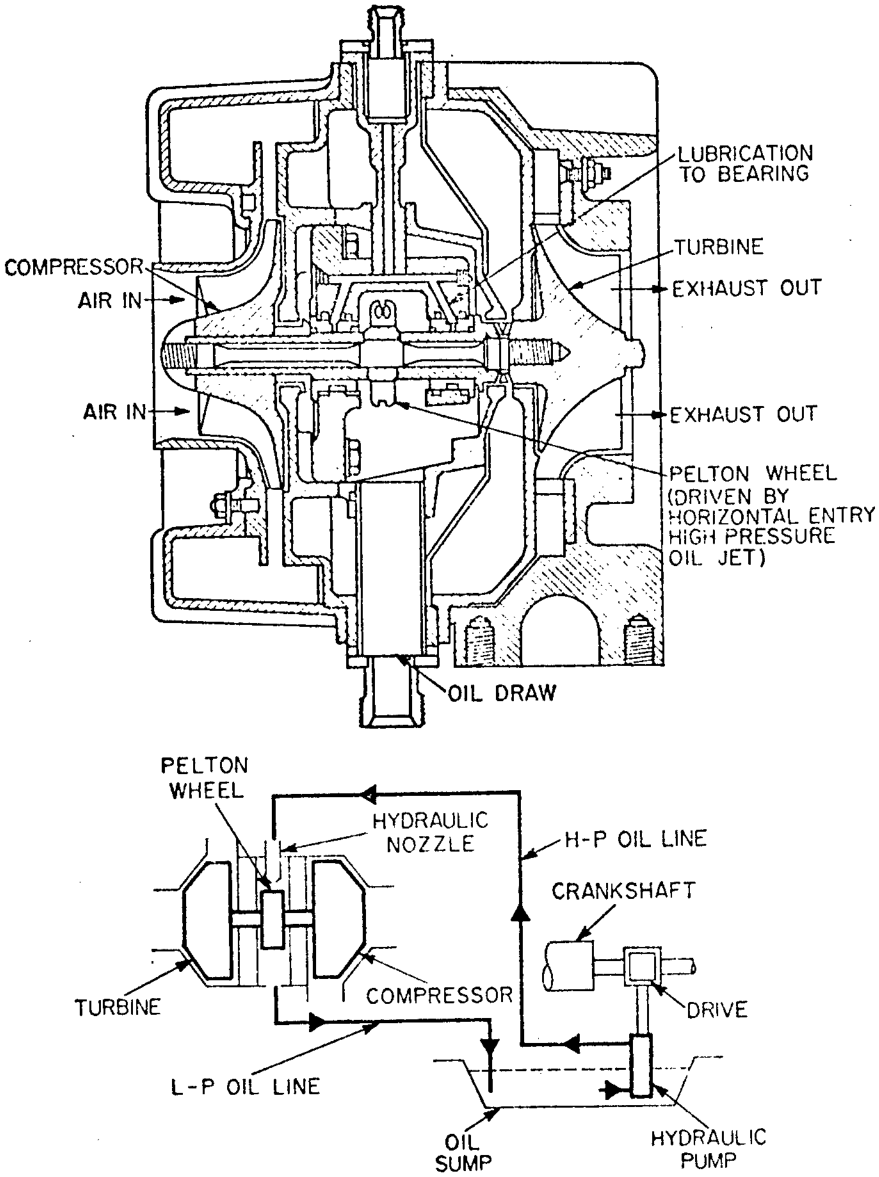


FIGURE 3.2.2 · Conceptual Pelton Wheel and Hydraulic Pump Supplemental Drive System for High Speed Diesel Engines (Ref. 12).

is increased by 35% and the engine is operated at the fuel rate of point A, then the fuel/air ratio and smoke level is reduced. However, a significant increase in engine output is possible if the fuel rate is raised to keep the same fuel/air ratio as point A. Hence, the Pelton/turbocharger system offers the potential of lower smoke levels or increased engine power at low engine speeds. With the Pelton wheel supplement and variable geometry compressor surge controls the response characteristics and the torque of the VHO diesel engine at low engine speeds are greatly improved.

The Pelton turbine lends itself to the turbocharger application because it operates quite efficiently at the same shaft speed as the turbocharger without the necessity of gearing (Figure 3.2.3). The wheel tip diameter for a turbine sized to the VHO diesel turbocharger is only about 2 inches, and the hydraulic pump supplies roughly 2000 psi oil pressure to the turbine at a rate of about 2.5 lb/sec (20 gpm). Obviously the Pelton system adds a great deal of complexity and cost to the turbocharging system of conventional diesel engines. Relatively high reliability could be expected from the individual components (including the low tip speed turbine wheel and the proven hydraulic components of the Pelton system), but the complexity of the system would probably reduce the overall reliability to a level comparable to exhaust waste gate devices. A typical system would be sized to provide supplemental power of about 15-20 HP to the turbocharger shaft at engine speeds between idle and 1800 RPM. With this increase in compressor power the boost of the VHO turbocharger (with variable diffuser compressor) could be increased from 38 to 53 inches HG Abs. at 1200 RPM and from 53 to 62 inches Hg Abs. at 1600 RPM. At higher speeds (above 1800 RPM) enough engine exhaust energy is available to drive the compressor to high boost levels without assist, so a control must be devised which shuts down the hydraulic system during high speed operation.

The performance of an engine with the Pelton boost augmentation system is shown in Figure 3.2.4. With the Pelton system it was possible to keep almost constant torque from 2600 to 1000 RPM without excessive

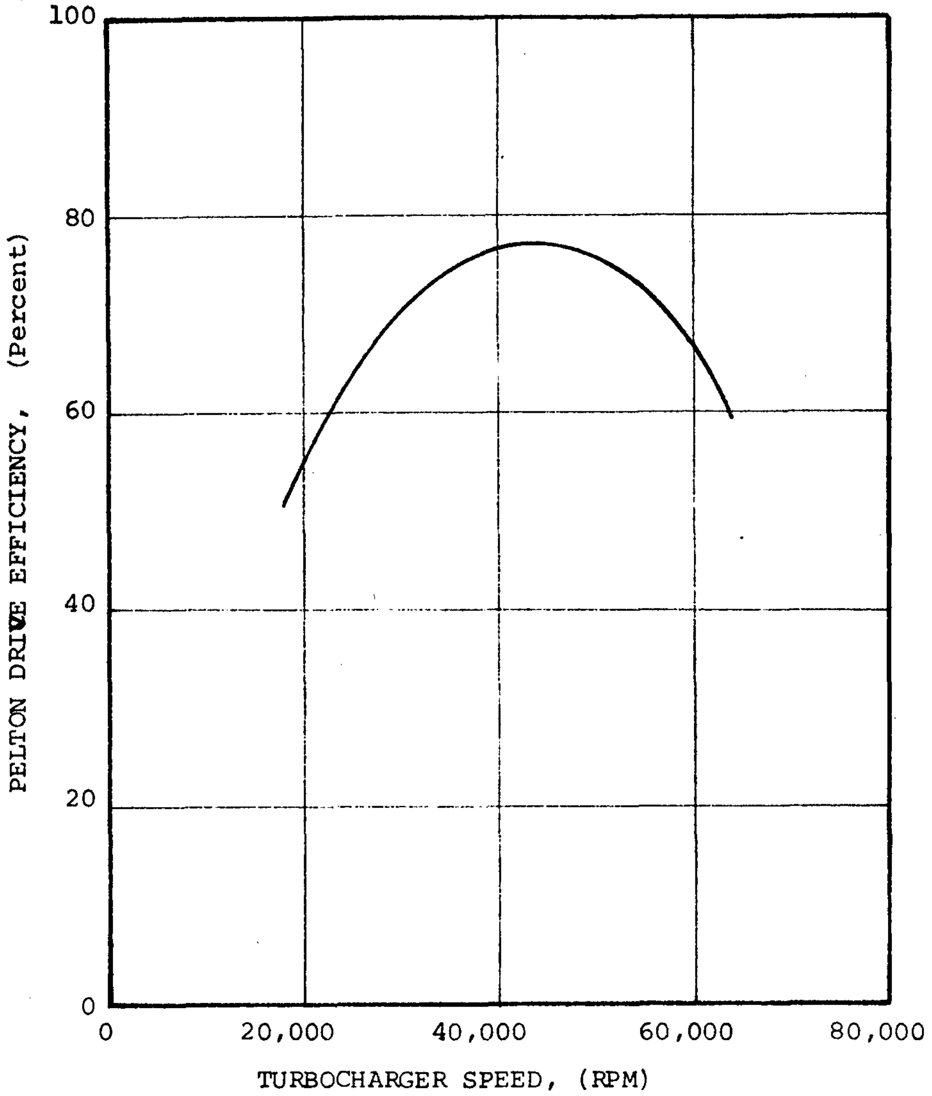


FIGURE 3.2.3 Estimated Pelton Wheel Drive Efficiency (Ref. 12).

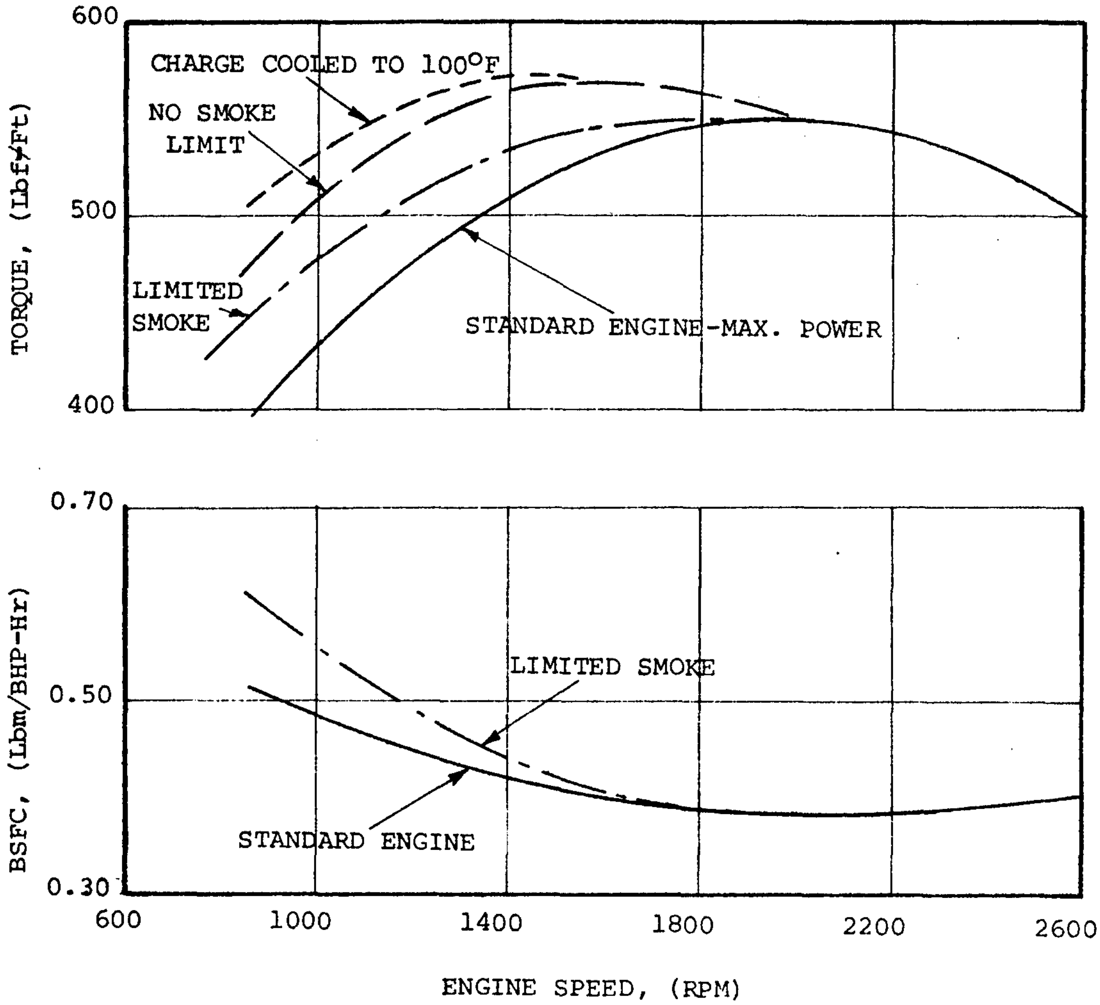


FIGURE 3.2.4. Part Speed Power Improvement with Pelton Drive Supplement.

smoking. By ignoring smoke limitations and operating at the fuel/air ratios and smoke intensity of the standard engine, about 5% higher torque was achieved. In the limited smoke tests about 11% higher fuel consumption resulted at 1000 RPM compared to the standard engine, but no penalty was involved at higher speeds.

Compared with other proposed means of transfering power from the engine shaft to the turbo shaft, all of which add complexity to the relatively simple turbocharger system, the Pelton system is superior. A variable diameter V-pulley system promises to be heavy, bulky, and not very reliable. Fluid coupling systems are inherently simpler, but cost of the 30:1 gear reduction machinery is a disadvantage.

The Pelton system has a slight performance advantage compared to the exhaust waste gate systems now used on a large number of turbocharged diesels. In the waste gate system additional turbine back pressure is put on the engine to boost power at low engine speeds. At higher speeds, where engine exhaust energy is greater, exhaust gases must be bypassed to avoid overboosting the engine. Additional back pressure on the engine leads to slightly lower efficiency with the waste gate system. The advantage of the Pelton system is that at high speeds the system is inoperative and has no effect upon engine efficiency. Over the life of a diesel engine this could amount to a considerable fuel saving, especially in typical truck applications where most of the fuel is expended at high power output. It is not obvious, however, that the potential fuel saving would recommend the additional cost and complexity of the Pelton drive supplemental boost system.

Exhaust energy increase can also be achieved with a variable turbine nozzle. For conventional radial turbines this device consists of moveable nozzle vanes similar to variable compressor diffuser hardware. With variable turbine geometry it is possible to obtain large boost increases at lower engine speeds without compromising high speed performance. Since the variable turbine nozzle allows higher turbine efficiency to be obtained over the broad spectrum of engine operating conditions, it actually improves engine efficiency by reducing the engine back pressure. However, the variable turbine nozzle concept is relatively unproven in operation and may prove to be a difficult device to maintain in the exhaust environment in which it must operate.

Another means of increasing the exhaust energy of turbocharged diesel engines under lug conditions is to utilize a blowdown turbine to extract the power needed for inlet air compression. Compared to a steady flow turbine, the blowdown turbine allows each element of exhaust gas to be utilized at its highest pressure and temperature. For example, at the early part of the cylinder blowdown process, the pressure and temperature of the discharging gas are high. Naturally, more available energy can be obtained from this high energy gas than if it were allowed to mix with later low pressure, low temperature portions of exhaust gas. Schweitzer and Tsu (Ref. 13) have shown that for a typical example an ideal steady flow turbine (one which utilizes a large receiver to collect exhaust gases upstream of the turbine) can provide only 58.5% of the theoretical exhaust gas utilization of the blowdown turbine. Development and testing of a blowdown turbine for the Army VHO diesel engine is currently being conducted under two Army programs (Contracts DAAE07-72-C-0212 and DAAE07-73-C-0147). One of the goals of these programs is to experimentally determine whether complex boost augmentation devices such as variable turbine geometry, Pelton wheels, etc. can be eliminated through efficient utilization of the potential of the blowdown turbocharging process.

In summary, it is concluded that for achieving increased lug capability of conventional turbocharged diesels (where modest boost increases are required), the Pelton wheel supplemental boost system probably will not replace the conventional waste gate device. The slight advantage of the Pelton drive system in engine efficiency at full load does not appear to warrant a large increase in turbocharger system complexity and cost. For advanced engine applications such as the variable compression diesel (which can use nearly constant boost at all engine speeds), the level of boost augmentation required demands exhaust utilization beyond the capability of the Pelton and waste gate systems. Such applications require more effective exhaust gas utilization that only the blowdown turbocharger or variable turbine hardware can provide.

3.3 The Effect of Valve Timing on Turbocharger Performance

An analysis has been made to ascertain the effects of valve timing on turbocharger compressor surge characteristics and turbine exhaust energy recovery. The purpose of this analysis was to evaluate the use of valve overlap scavenging on high speed turbocharged diesels as a method of improving part speed performance and reducing smoke while maintaining acceptable high speed performance.

Two 4-stroke diesels which exhibit considerable difference in valve timing events were studied:

1. The Caterpillar VHO (525 CID) engine.
2. The Continental AVCR-1360 variable compression ratio (1360 CID) engine.

Both of these engines are highly turbocharged and require high exhaust back pressure to give the necessary turbine power to drive the compressor. Table I compares the valve characteristics of the two engines. As shown, the intake and exhaust valve areas and the valve overlap period are considerably larger for the AVCR-1360 engine than for the VHO engine. In fact, the intake area is about 14% larger, the exhaust area about 30% larger, and the valve overlap period over 50% larger. When the valve areas as listed in Table I are combined with the experimentally determined flow coefficients, the effective valve area curves as a function of crank angle are obtained, as shown in Figure 3.3.1. These curves dramatically illustrate the significantly larger valve overlap flow area of the AVCR-1360 engine.

The valve overlap profiles shown in Figure 3.3.1 have significant influences on the compressor surge and smoke characteristics, especially at low engine speeds. This is due to the fact that the larger valve overlap increases the engine airflow, thereby shifting the constant engine speed operating lines to higher airflow and to the right of the compressor surge line. At low engine speeds, where the maximum boost is often surge limited, this increased airflow will allow higher boost pressures, which in turn will permit increased engine fuel flows without smoking.

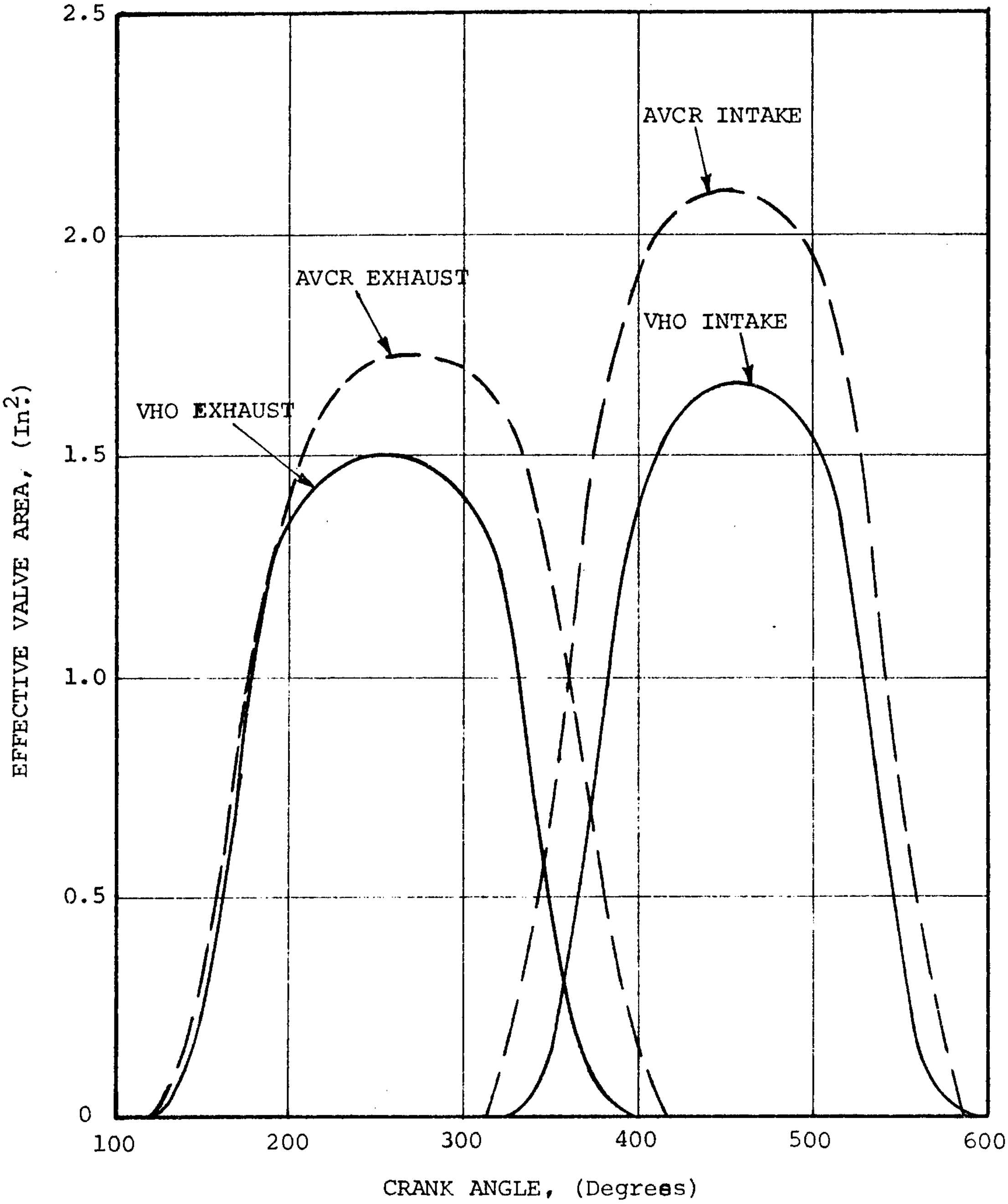


FIGURE 3.3.1. Effective Intake and Exhaust Valve Flow Areas for the VHO and AVCR-1360 Diesel Engines.

These compressor/engine matching characteristics are illustrated in Figure 3.3.2 and 3.3.3, the compressor maps for the VHO and AVCR-1360 engines.

Figure 3.3.2 indicates that the VHO engine is surge limited to a boost pressure ratio of only about 1.6 at 1600 RPM while Figure 3.3.3 indicates that the AVCR-1360 engine is surge limited to a boost pressure ratio of about 2.6 at this same speed. There are, of course, differences in the two compressor maps which affect a surge margin comparison; however, a considerable degree of this surge margin difference and increased pressure ratio is directly attributable to the increased airflow (blowthrough) during the valve overlap period on the AVCR-1360 engine. In effect, this large valve overlap period of the AVCR-1360 engine is acting as a "blowoff valve" on the compressor discharge, a method for increasing compressor surge margin discussed earlier in section 3.1.3 of this report. But this surge control method is more efficient because the excess airflow is subsequently expanded in the exhaust turbine to increase available exhaust energy.

An important point to note concerning the blowthrough of both the VHO and AVCR-1360 engines is the fact that the percentage of blowthrough increases as the engine speeds decreases. This is quite desirable since more surge margin and therefore blowthrough is needed at the lower engine speeds. In addition, since surge control is not needed at high engine speeds, it is desirable to reduce the blowthrough, since this blowthrough increases the increased compressor power requirement and requires an increased engine back pressure (or higher f/a ratio). Table II shows the air flow rates of the VHO and AVCR-1360 engines at rated engine speeds and at 1600 RPM. These results were generated using the TMSCO exhaust energy recovery computer program. For the same boost level at design speed and at 1600 RPM one would expect the engine air flow to be directly proportional to engine speed if the percentage of blowthrough remained constant. As Table II indicates, however, the blowthrough increases by about 5% for the VHO engine and by about 12%

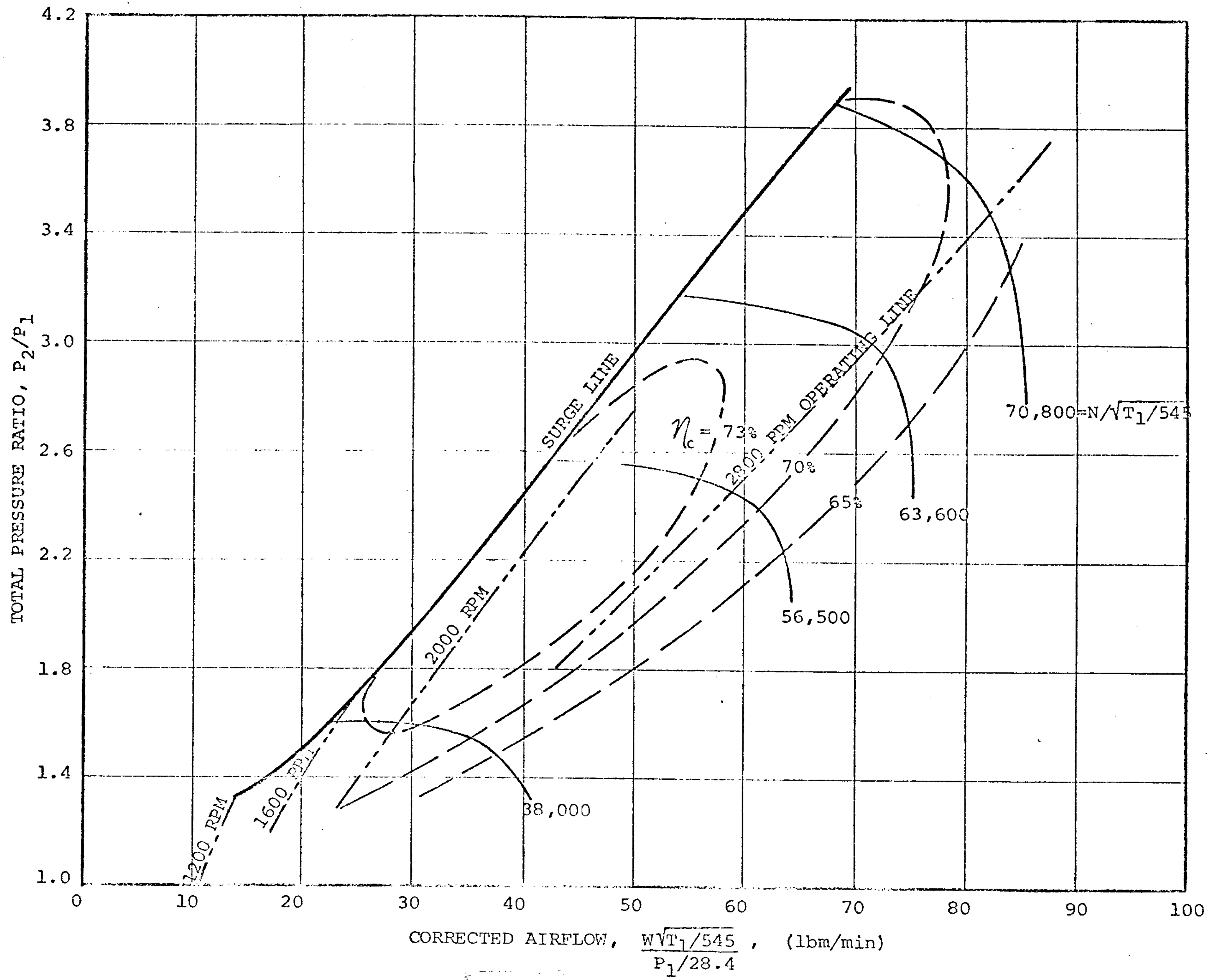


FIGURE 3.3.2 . Standard Compressor Map with VHO Operating Line Superimposed.

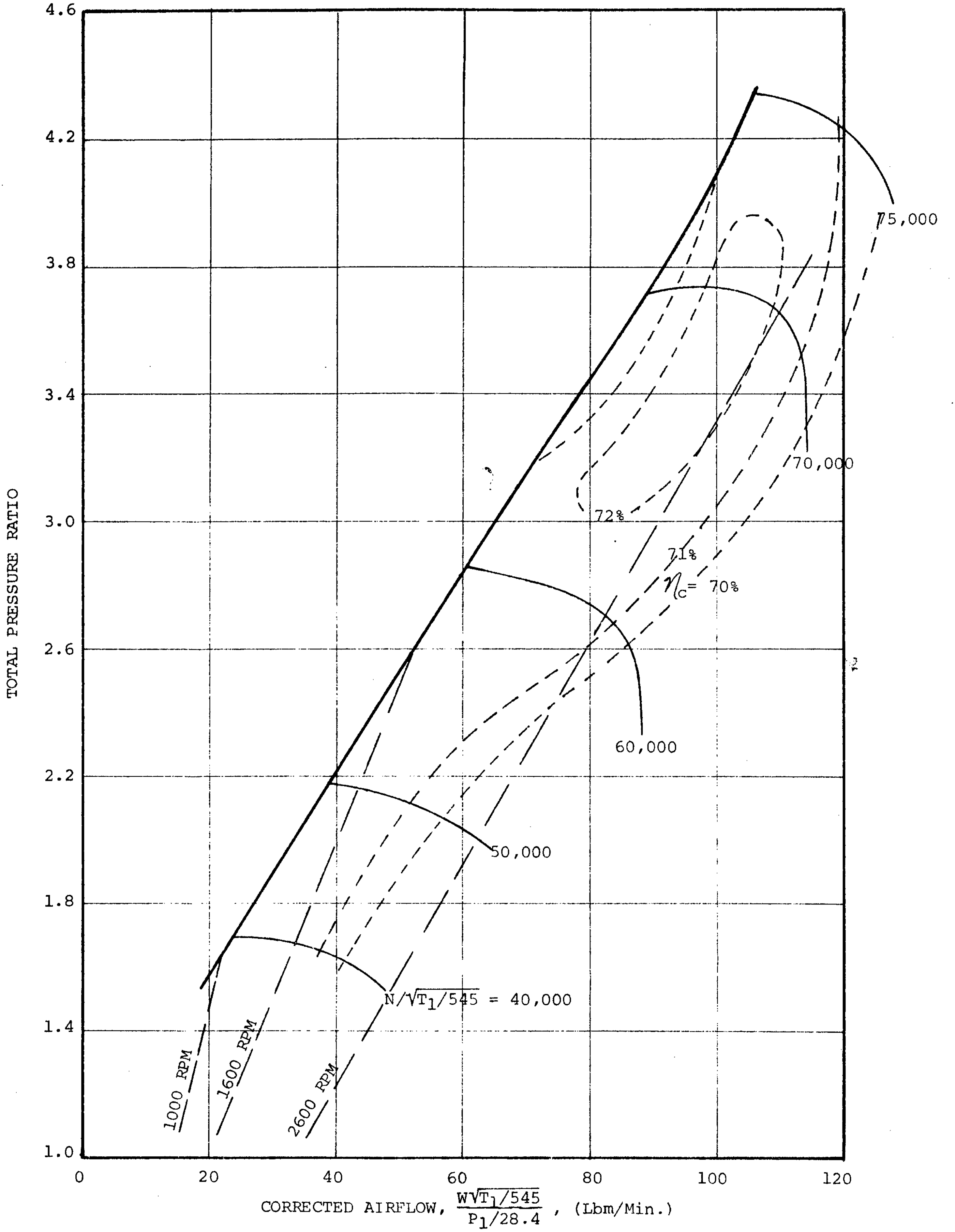


FIGURE 3.3.3. AVCR-1360 Compressor Map with Operating Lines Superimposed.

for the AVCR-1360 engine at 1600 RPM. The reason for the increased blowthrough is simply that at 1600 RPM, the actual time during the valve overlap period is greater than that at rated speed (by the factor, $\frac{\text{rated speed}}{1600}$).

TABLE II

EFFECTS OF ENGINE SPEED ON BLOWTHROUGH FOR THE VHO AND AVCR-1360 ENGINES.

ENGINE	BOOST PRESSURE ("Hg)	AIR FLOW *RATED RPM (lb/min/6cyl)	AIR FLOW 1600 RPM (lb/min/6cyl)	% BLOWTHROUGH INCREASE AT 1600 RPM
VHO	97	82.8	49.8	5%
AVCR-1360	130	134	92	12%

*Rated RPM; VHO = 2800, AVCR-1360 = 2600

Relatively large valve overlap has been shown useful in increasing compressor surge margin and boost at low engine speeds (thus allowing lower fuel/air ratios), which contributes to improved steady state performance and less smoke. However, during the transient or acceleration period, which is an especially important consideration in tank powerplants, the blowthrough obtained with high valve overlap tends to be counter-productive. That is, when rapid acceleration is required, opening of the throttle causes the exhaust back pressure to rise more rapidly than the compressor pressure, and a positive scavenge differential cannot easily be maintained. Rather than having fresh air blowing through the cylinder, exhaust gas can blow back into the combustion chamber and interfere with the combustion process. The end result is slower acceleration and more smoke during the acceleration period. Most high speed vehicle engines subject to rapid load changes, then, will not necessarily be able to use valve overlap effectively (the AVCR-1360 engine is an exception). The variable compression ratio feature of the AVCR-1360, which allows compression ratios to be reduced at high power conditions, results in a higher exhaust temperature than a conventional diesel. Thus, the exhaust energy is greater and less exhaust back pressure is required to achieve a given boost pressure (a larger positive differential is maintained than in conventional diesels). This characteristic allows the scavenge differential during the rapid acceleration period to remain somewhat more favorable than would be possible with a high compression ratio diesel cycle.

In conclusion, it appears that increased valve overlap does provide a reasonable method for providing modest increases in surge margin of turbocharger compressors, thereby allowing higher boost at low speed and lower fuel/air ratios for a given fuel rate. To reduce smoke and obtain acceptable response to acceleration demands, it is necessary to use appreciable valve overlap only when a reasonable scavenge differential can be maintained during both steady state and transient operation (i.e., limit it to variable compression ratio engines or develop much more efficient turbocharger components). The degree of valve overlap should be dictated by the desired surge margin improvement and reasonable engine back pressure limitations.

4.0 EXPERIMENTAL PROGRAM

4.1 Design of Variable Diffuser Turbocharger Compressor

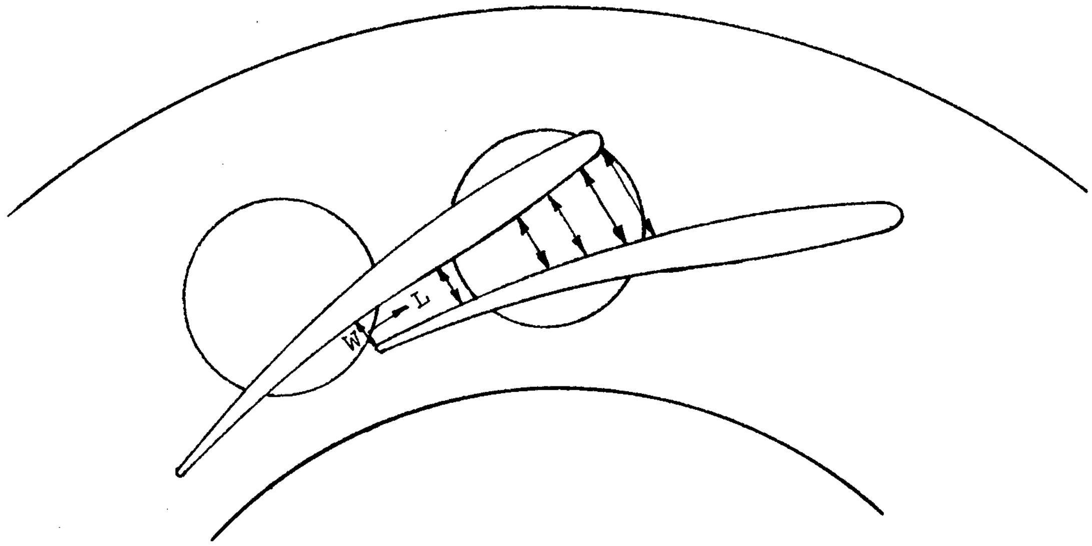
The results of the analysis conducted in this program indicated that the most practical variable geometry surge control method for turbocharger compressors is the variable vaned diffuser. Various types of variable diffuser designs were studied, as well as available compressors suitable for design and fabrication of a variable geometry turbocharger compressor for the VHO diesel. This work led to the selection of the VHO turbocharger compressor for variable diffuser modification.

Certain characteristics of the standard VHO compressor made it attractive for this application. Of course, the compressor was sized and matched to the VHO engine boost and airflow requirements. The impeller geometry for this turbocharger compressor was optimized for high stability, although perhaps typical of turbocharger compressors in that peak efficiency was 5-10 points lower than a design optimized for maximum efficiency. The high impeller diameter ratio ($\epsilon = 2.1$) of the VHO compressor helps to unload the inducer by requiring less incidence change for a given change of flow rate. Therefore, at reduced throat area settings this compressor promised to exhibit slightly less impeller stalling tendency and thus reduced efficiency decay compared to the Solar variable geometry compressor ($\epsilon = 1.96$) reported in Ref. 5. Also, the performance of the VHO compressor in the design vane configuration was known, so the effect of variable geometry modifications could be easily determined. Another factor in selection of the VHO compressor for variable diffuser geometry was the relative simplicity of the modification procedures for an existing turbocharger compared to the problem of incorporating a different compressor into a compressor/ turbine rotor dynamic system for which it was not originally intended. By utilizing the VHO compressor and the standard turbocharger bearing and turbine assembly no changes are required to the rotating components (rotor, shaft, and bearings); the essential modifications for variable diffuser geometry are in general limited to the diffuser system itself.

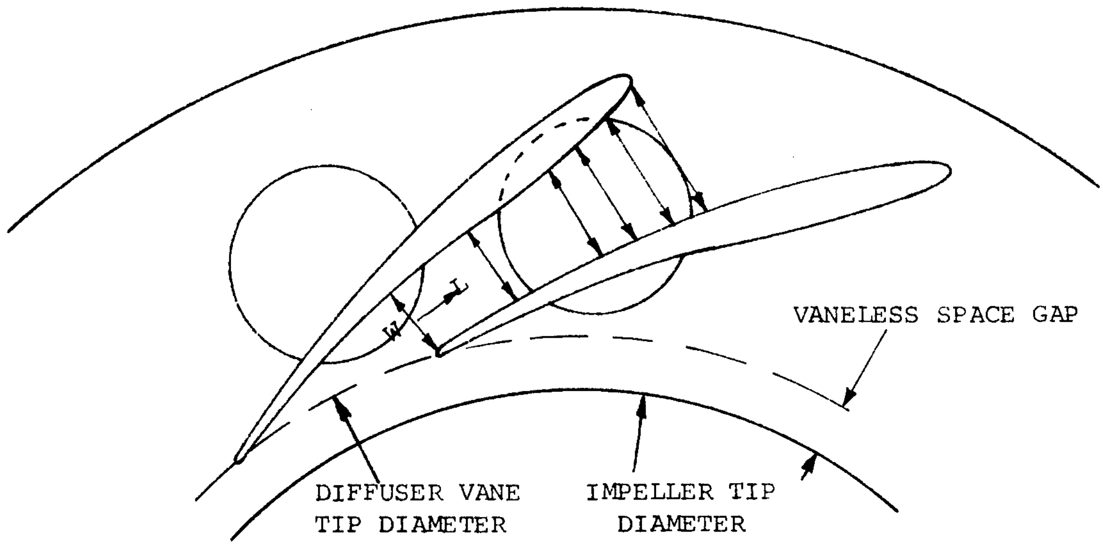
Variable diffuser modification required that the standard fixed vanes be machined off and replaced with vanes attached to a rotating pivot shaft passing through the rear diffuser wall. Utilization of the existing vane airfoil profile was considered because the performance of the compressor would be known in the design setting. A sketch of the standard diffuser vanes in both the design (100%) throat area position and the 45% position is shown in Figure 4.1.1. The 45% setting is expected to be typical of the maximum adjustment necessary to achieve the desired range improvement for typical diesel turbocharger applications. It also corresponds to a reasonable flow reduction limit for the variable diffuser compressor. More severe throat area adjustment than this was expected to cause impeller stalling and heavy recirculation of heated, compressed air into the inlet. Under these conditions the compressor exhibits significantly reduced efficiency and pressure rise potential.

In order to utilize the existing diffuser vane profile the channel divergence had to be checked in both the design and 45% setting. In the 45% setting the diffuser channel area ratio is increased to 2.8 from 1.8 in the design setting. These geometries are spotted on the familiar Kline (Ref. 14) two-dimensional diffuser performance curve illustrated in Figure 4.1.2; both geometries are shown to fall below the line of appreciable stall. If stalling is exhibited in the more critical 45% setting, it is probable that it will occur in the last 10% of channel length after most of the necessary diffusion has been accomplished. Therefore, the aerodynamics of the standard vane profiles were considered acceptable and a variable diffuser compressor design was generated from this basic diffuser geometry.

From Figure 4.1.1 it can also be observed that the vanes are pivoted about a point nearer the vane leading edge (also near the aerodynamic center where minimum torque is required to actuate the vanes) and are attached off center on the pivot shaft disc. This vane pivot location was



45% THROAT AREA POSITION



DESIGN VANE POSITION

FIGURE 4.1.1. Variable Vaned V/to Diffuser Showing Divergence of Channel in Design Throat Area Position and 45% Throat Area Position.

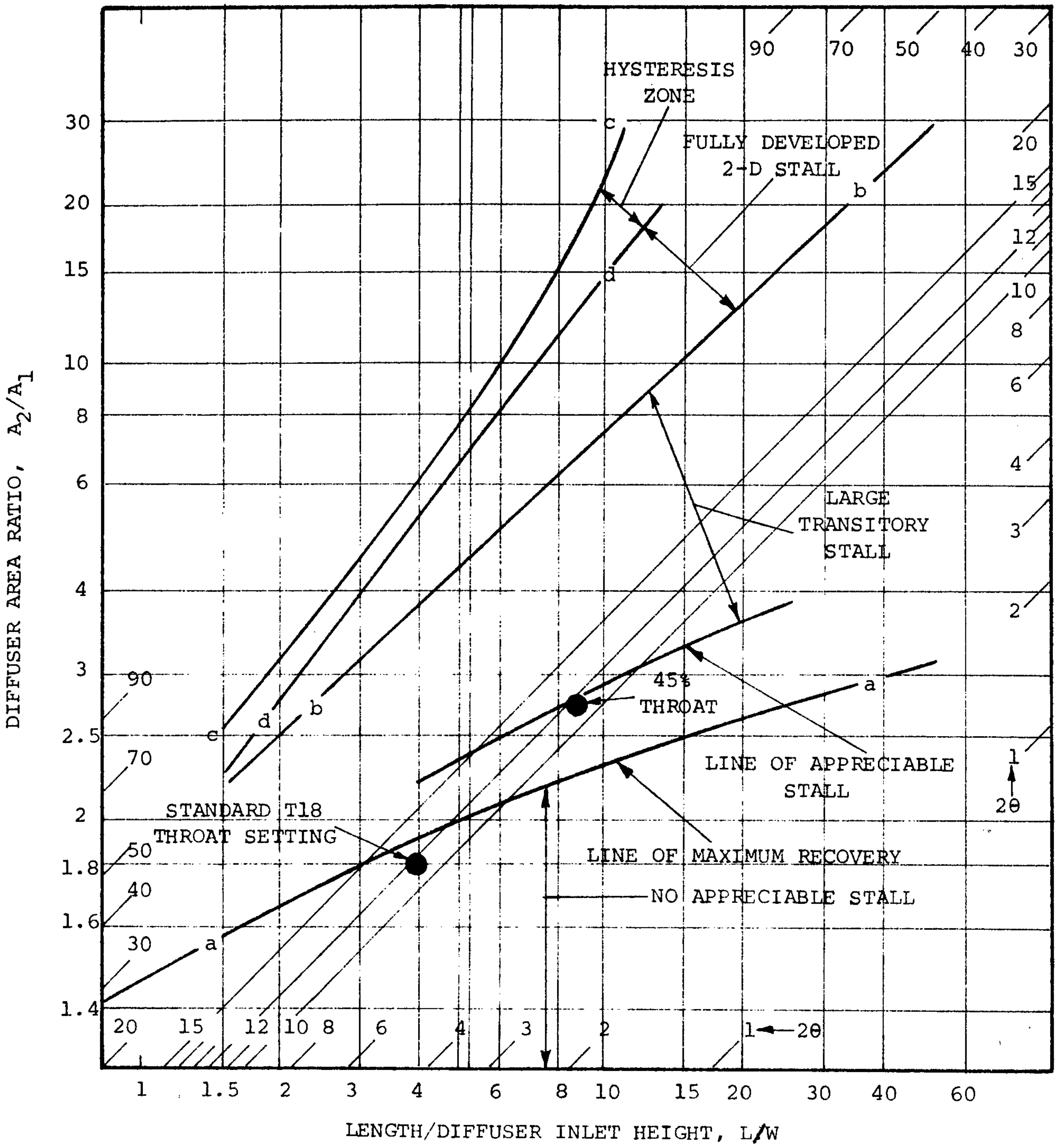


FIGURE 4.1.2 Kline Two-Dimensional Diffuser Performance Curve Showing Geometry of VHO Variable Diffuser in Design and 45% Throat Area Settings (Ref. 14).

chosen to allow rotation to the extreme closed settings with minimum change in vaneless space ratio (vane tip diameter/impeller tip diameter). By minimizing the vaneless space gap it is possible to reduce the excess friction losses which occur at low flow rates when the spiral path of air moving from the impeller to the diffuser tip increases significantly. Therefore, the efficiency potential of the variable diffuser compressor at partial vane settings is increased relative to a design such as Solar utilizes which did not attempt to control the vaneless space gap.

Other than the previously described design changes in the diffuser there were no aerodynamic design deviations from the standard geometry. Compressor and turbine performance of the standard turbocharger were known, so the effect of the variable diffuser apparatus was to be evidenced by the measured performance of the variable geometry turbocharger in the design setting. Off-design performance was expected to be comparable to results achieved in the Solar tests (Ref. 5) with a radial compressor of similar geometry, although a slight improvement in the area of efficiency decay was possible because of a slightly more stable impeller geometry and improved variable diffuser design. An estimated off design performance map for the variable diffuser compressor was constructed from the Solar variable geometry results (Figure 4.1.3). Flow rate was estimated to be proportional to throat area adjustment. Efficiency and pressure ratio decay comparable to the Solar tests are shown.

Two types of variable diffuser turbocharger compressors were designed for this program. The basic diffuser aerodynamic design features described above were applied to each design. The primary difference was in the actuation system for the vanes.

A layout drawing of a conceptual turbocharger utilizing a gear type actuation system is shown in Figure 4.1.4. The fixed diffuser vanes were removed from the outer diffuser wall and replaced with adjustable vanes of the same airfoil profile and incidence. Each of the new vanes is integral with a pivot shaft which passes through the inner diffuser wall

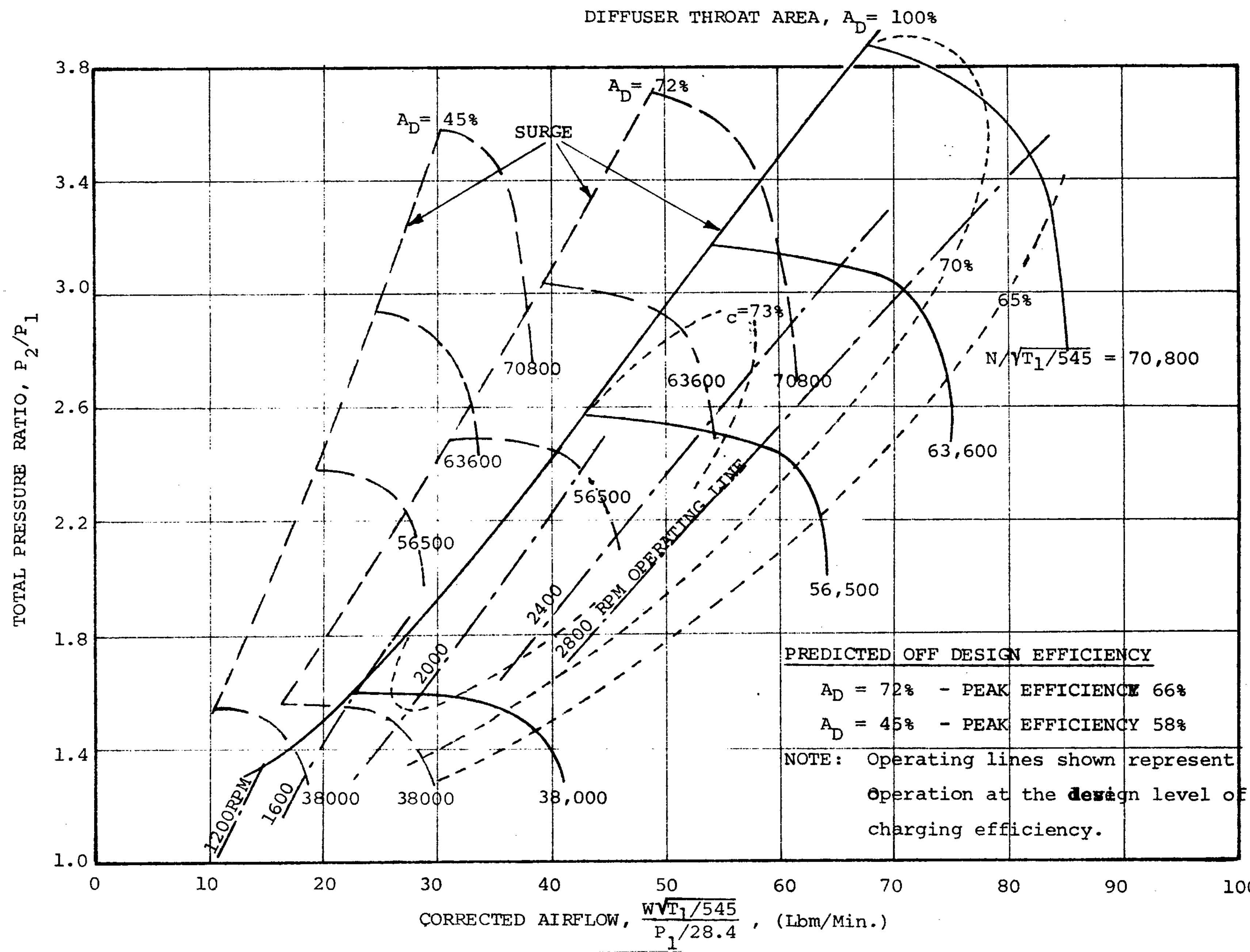
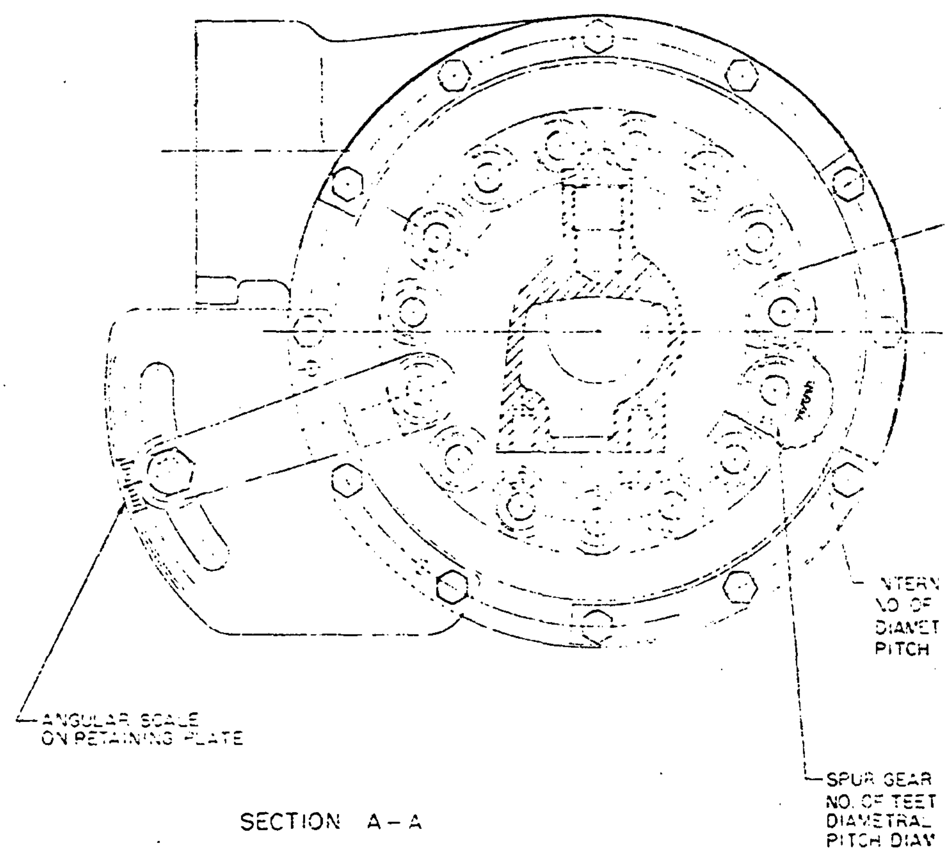
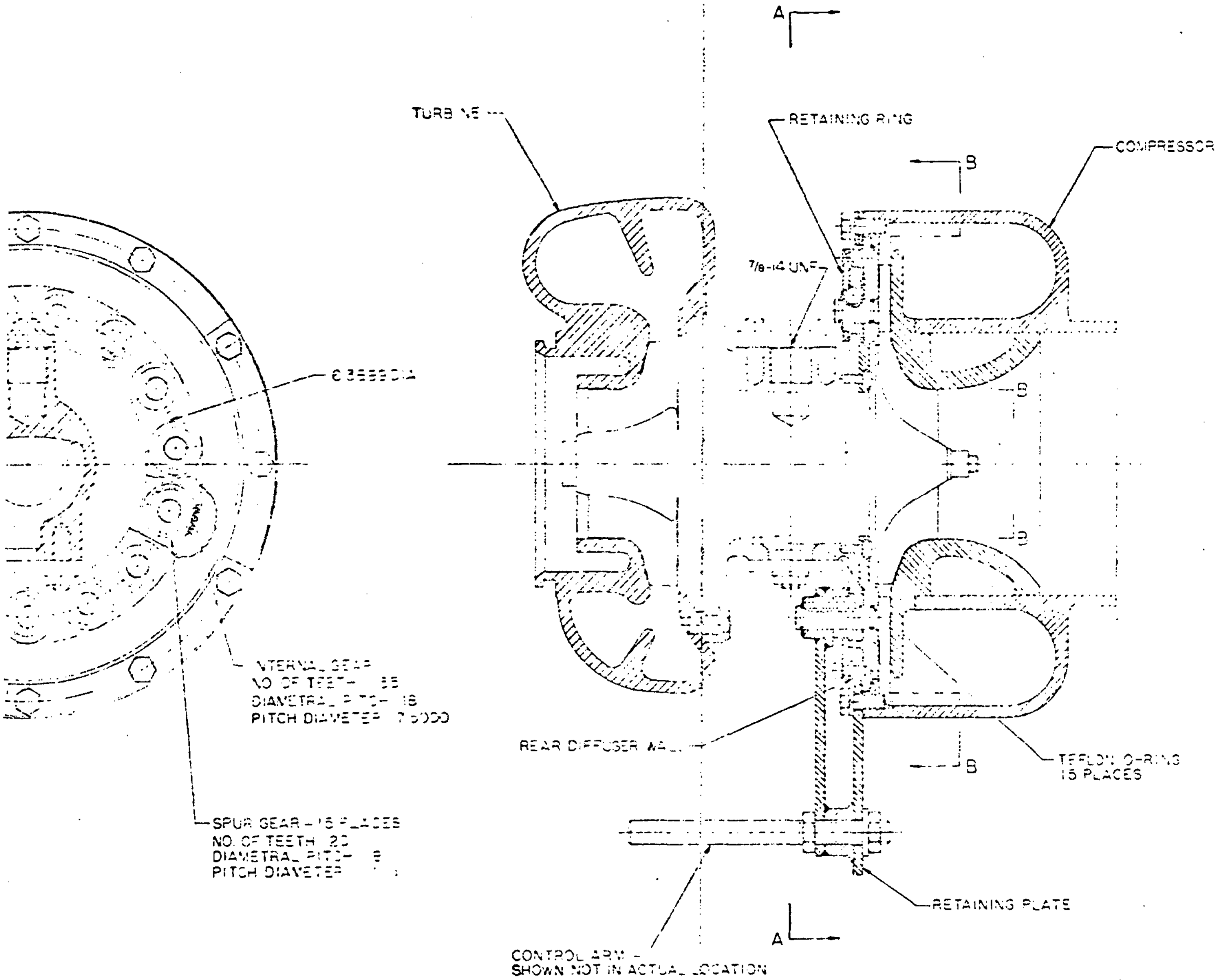


FIGURE 4.1.3. Estimated Performance of Compressor with Adjustable Diffuser vanes.


ANGULAR SCALE
ON RETAINING PLATE
SPUR GEAR
NO. OF TEET
DIAMETRAL
PITCH DIAM
NTERN
NO. OF
DIAMET
PITCH

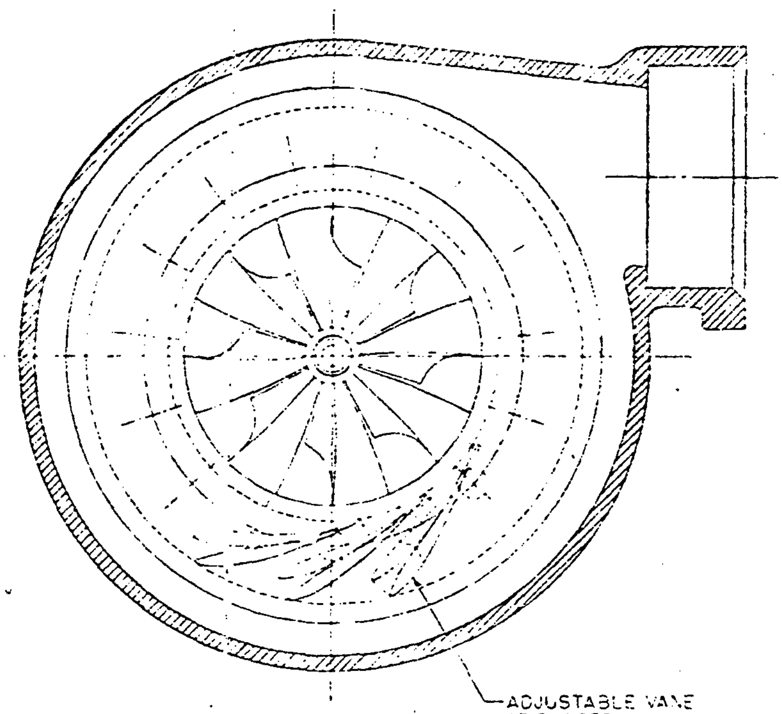
SECTION A – A

(1)


A
RETAINING RING
COMPRESSOR
B
7/8-14 UNF
INTERNAL GEAR
SPUR GEAR
NO. OF TEETH
DIAMETRAL PITCH
PITCH DIAMETER
TEFLON O-RING
15 PLACES
RETAINING PLATE
CONTROL ARM
SHOWN NOT IN ACTUAL LOCATION

2

—COMPRESSOR



SECTION B–B

ON O-RING
LACES

FIGURE 4.1.4

THERMO MECHANICAL SYSTEMS INC CANOGA PARK, CALIF			
SCALE FULL	APPROVED		DRAWN BY L N
DATE 9-22-72			REVISED
VARIABLE GEOMETRY TURBOCOMPRESSOR			
525 CID VHC F7X2528 —			DRAWING NO LO-58R

3

and is connected to a spur gear. A lever arm attached to one of the 15 vanes transmits rotational motion to all of the vanes through a large internal gear. The spur gears are locked in position on the pivot shaft with Woodruff keys. Approximately 30° of rotation of the lever results in a change of throat area from the design position to the almost fully closed position shown.

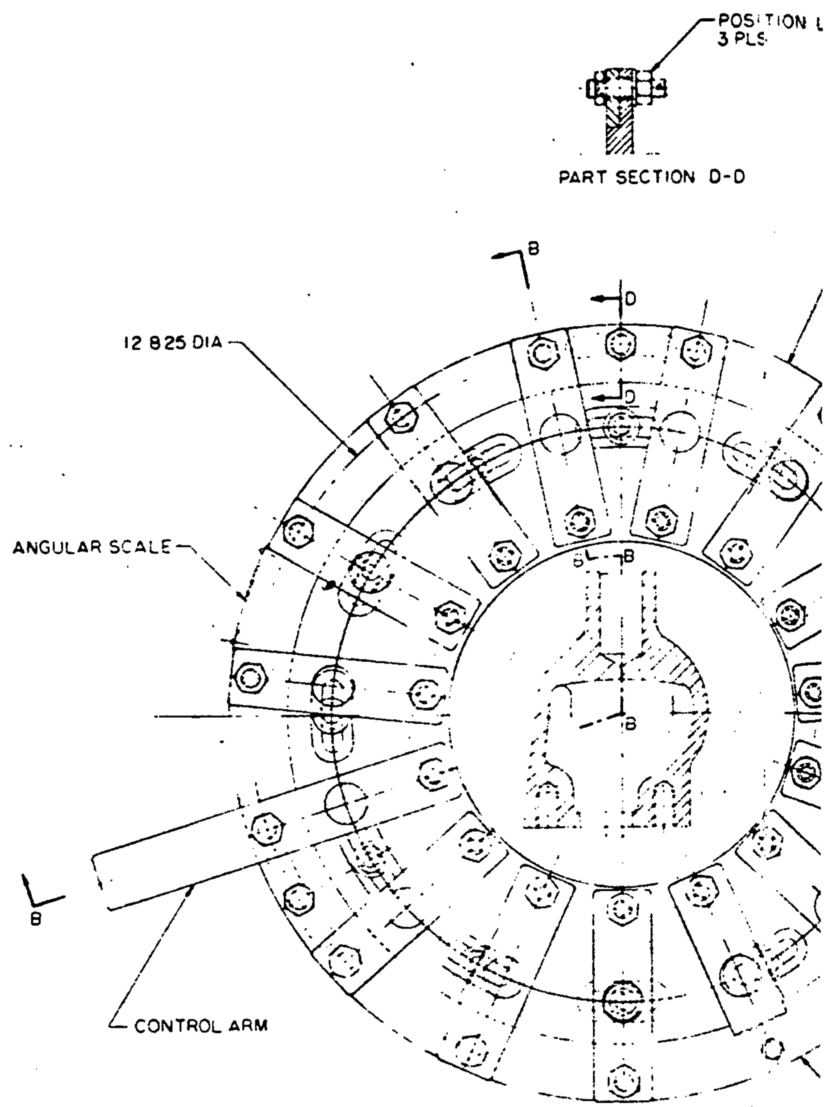
Critical examination of this design indicated several potential problem areas, namely:

1. The large internal gear to the fifteen pinions must be in exact alignment and there is no easy way to adjust for the tolerance build up.
2. The use of Woodruff keys would pose a difficult problem in alignment of the spur gears on the fifteen pivot shafts to obtain consistent vane airfoil positioning circumferentially.

Because of these potential problems, a new adjustable diffuser vane layout was initiated.

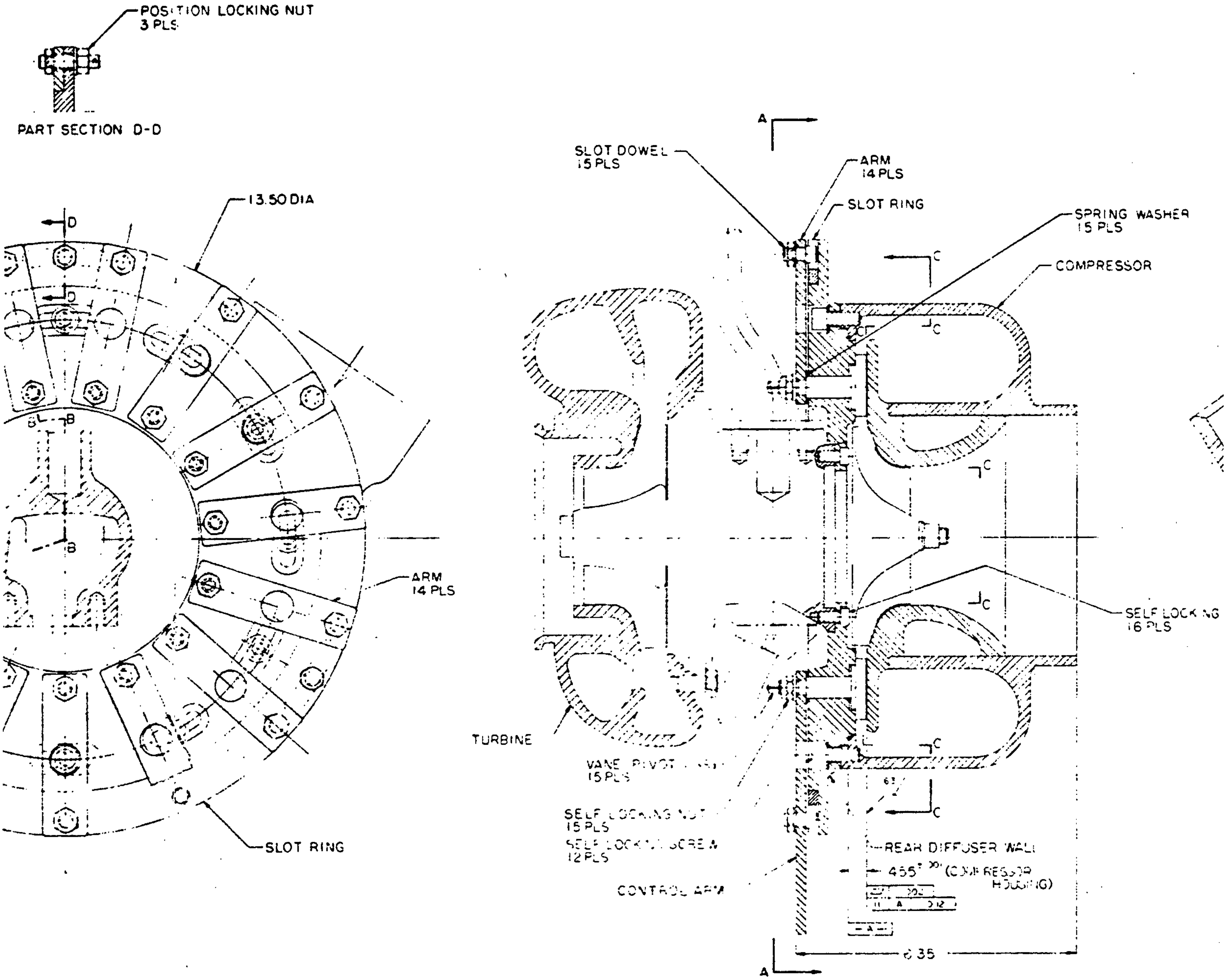
In the second design an actuation arm and ring assembly (Figure 4.1.5) replaced the gear actuation system. Again the 15 fixed vanes were removed and replaced with identical vane airfoil profiles integrally machined with the pivot shafts from individual billets. Close tolerances were maintained on the airfoil profiles and the height of the pivot shaft disc (to eliminate abrupt steps on the diffuser channel surface). Each pivot shaft passes through and is supported by the rear diffuser wall. The pivot shafts are shown bolted to a metal ring which is firmly supported by a flange in the rear diffuser wall housing. The actuation ring rides on the flange and the contact surface is dry lubricated by a graphite film (graphilm, a product of Graphalloy Metallizing Corp.). This lubrication method should allow ease of movement of the ring without need for expensive bearings, particularly in the prototype hardware.

The outer diameter of the ring is slotted vertically at fifteen equi-distant locations and bolts from the outer lever arm ride in these slots. Close tolerances must be held circumferentially but not vertically


POSITION L
3 PLS
PART SECTION D-D
B
D
12 825 DIA
ANGULAR SCALE
CONTROL ARM

SECTION A-A

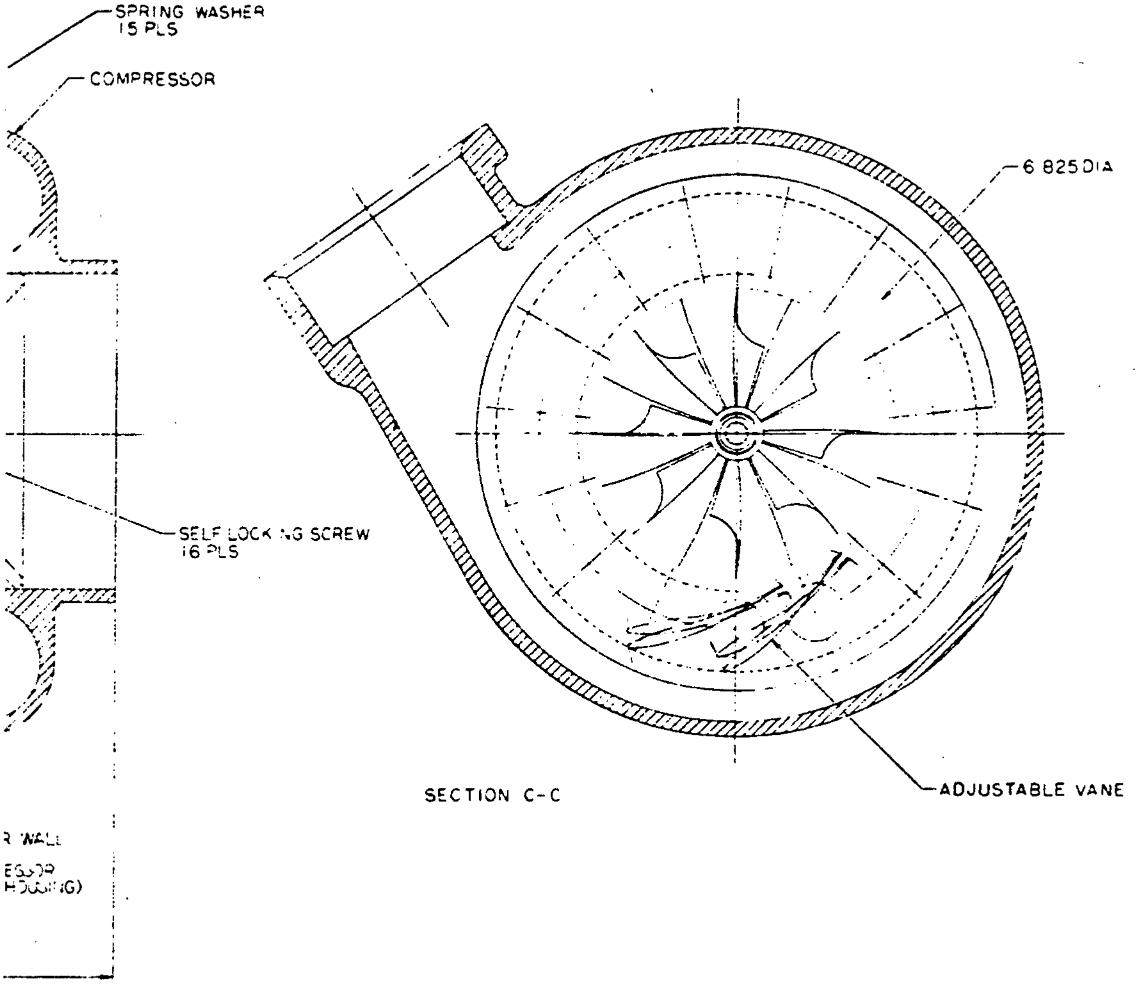
(1)


POSITION LOCKING NUT
3 PLS
PART SECTION D-D
13.50 DIA
ARM
14 PLS
SLOT RING
SLOT DOWEL
15 PLS
ARM
14 PLS
SLOT RING
SPRING WASHER
15 PLS
COMPRESSOR
SELF LOCKING
16 PLS
TURBINE
15 PLS
REAR DIFFUSER WALL
(COMPRESSOR HOUSING)
CONTROL ARM
6.35

SECTION B-B

2

LTR	DESCRIPTION	DATE	APPROVED
A	SLOTS IN REAR DIFFUSER WALL, WERE HOLES	11-20-72	
B	.455 ±.001, WAS .458 ±.001 RECTANGULAR ARMS, WERE CHAMFERED	12-20-72	



SECTION C-C

3

FIGURE 4.1.5

THERMO MECHANICAL SYSTEMS INC CANOGA PARK, CALIF			
SCALE FULL	APPROVED	DRAWN BY L N	
DATE 11-15-72		REVISED 11-20-72	
VARIABLE GEOMETRY TURBOCOMPRESSOR – MOD A			
525 CID VHO F7X 2528 –		DRAWING NO 0211-1-B	

in the ring slots. Rotation of the ring by the control arm causes equal rotative movement for all lever arms and thus diffuser vanes. When the desired vane setting is achieved, the ring is securely fastened in place by tightening the three position locking nuts on bolts which ride in circumferential slots in the ring. These slots allow movement of the vanes from the fully opened to the completely closed vane setting, a total rotation of eighteen degrees of the pivot shafts. The system is designed such that a small movement of the ring circumferentially (0.005 in.) results in only a negligible rotation of the vane (5 minutes). This has been accomplished in a design which has only three locations where close tolerances are critical: the circumferential location of the 15 vertical slots in the actuation ring, the hole in the rear diffuser wall through which the pivot shaft passes, and the dimensions of the vane support disc which fit in the hole. No expensive bearings are required and in contrast to the geared actuation system there is no critical alignment problem other than setting the vanes accurately in the initial assembly of the turbocharger. This design is relatively simple and rugged and was selected as the preferred variable diffuser design for prototype fabrication and testing.

4.2 Compressor Fabrication and Assembly

Fabrication of the recommended variable diffuser design for the turbocharger commenced after verbal approval was granted by the USATACOM Project Engineer. Modification of the rotating components (rotor, shaft, and bearings) was not required and conversion to variable diffuser geometry was in general limited to the diffuser and the diffuser case. Two machining operations were necessary on the existing hardware. They consisted of removing the fixed geometry diffuser vanes and removing the forward bearing housing flange (rear diffuser wall). The following machined components of the variable diffuser system were fabricated:

1. Vane Setting Plate (one).
2. Rear Diffuser Wall (one).
3. Vane Pivot Shaft Assembly (15 pieces).
4. Slot Ring (one).
5. Vane Actuator Arms (14 pieces).
6. Control Arm (one).

All but the individual actuator arms were machined from 321 stainless steel; the arms were machined from 6061-T6 aluminum.

The vane setting plate (item 1 above) is not an integral component of the variable geometry system but was to be used to facilitate initial assembly of the variable diffuser vanes in the correct circumferential position. The hardware was fabricated by a local manufacturer, and no problems were experienced in assembly of the procured hardware. The variable geometry mechanism assembled easily and exhibited smooth operation on the first build. The photographs of the fabricated diffuser assembly (Figure 4.4.6 and 4.4.7) show the vanes in the design (100%) and closed (45%) throat area positions.

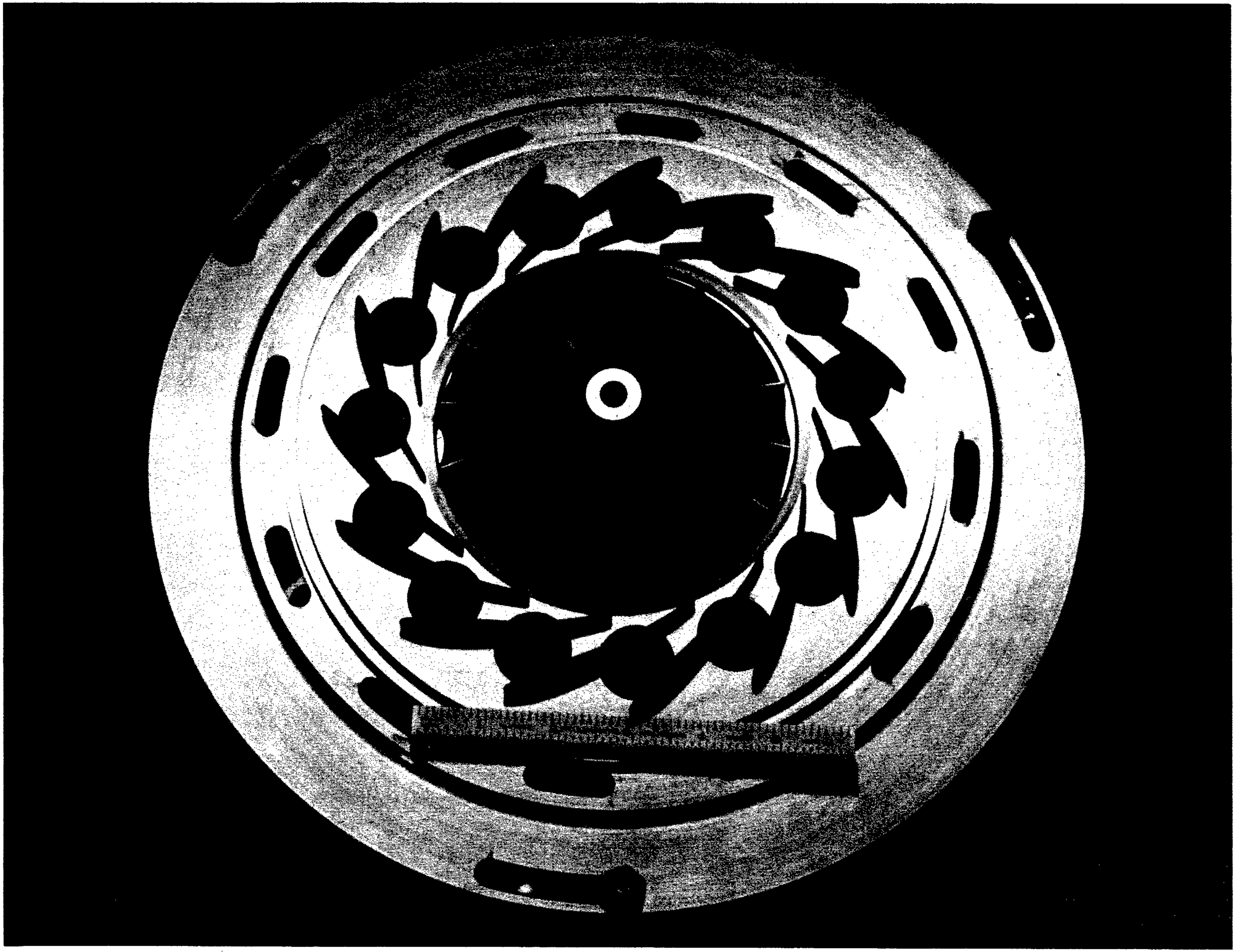


FIGURE 4.2.6: Variable Geometry Diffuser with Vanes in Design Position (100% throat area).

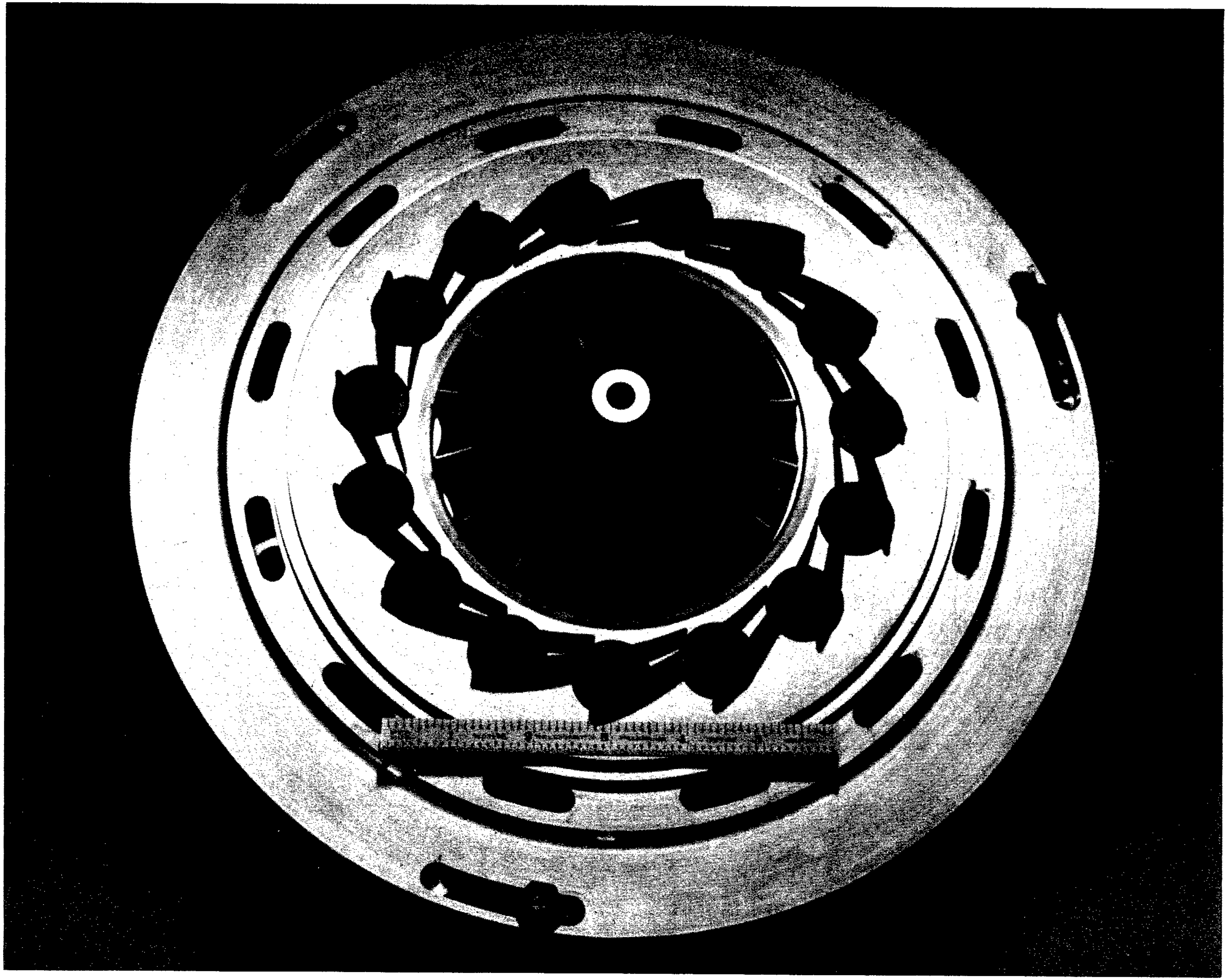


FIGURE 4.2.7: Variable Geometry Diffuser with Vanes in Closed Position (45% throat area).

4.3 Apparatus and Test Procedures

Other elements of compressor test rig hardware which were procured or fabricated for this program included:

1. Compressor inlet air metering device and ducting.
2. Compressor discharge throttling valve and associated ducting.
3. Turbocharger lube system.
4. Instrumentation.

The compressor test rig is shown assembled in Figure 4.3.1 prior to hookup of the facility turbine drive air supply.

The compressor inlet system consisted of a 30 inch length of 6 inch diameter ducting into which a flow measuring venturi was attached via a flange at the front face. Two venturi sizes were required to accurately measure the range of flow rates of the compressor. A smaller 1.792 in. diameter venturi was required for the airflow measurements up to 30 lb/min. Airflows between 30 lb/min and 100 lb/min require the larger 3.114 in. diameter venturi. A screen was located approximately midway in the inlet duct to help insure that a smooth, undistorted flow profile was obtained at the compressor face.

The compressor back pressure was controlled by a motor operated butterfly valve located in the compressor discharge duct. Air leaving the butterfly valve was discharged away from the test stand through an exhaust duct extension. The lubrication system consisted of an oil sump, pump, filter, and oil cooler.

Adequate instrumentation was provided to determine compressor performance and monitor the mechanical operation of the rig. The instrumentation is listed in Table III and a schematic diagram of the system showing the location of the various instrumentation items is provided in Figure 4.3.2. Automatic shutdown devices were provided to detect high oil outlet temperature (250°F) and low oil pressure (18 psig). A manual shutdown device was provided to shut off the turbine drive air supply.

The test rig was installed in the test facility at Approved

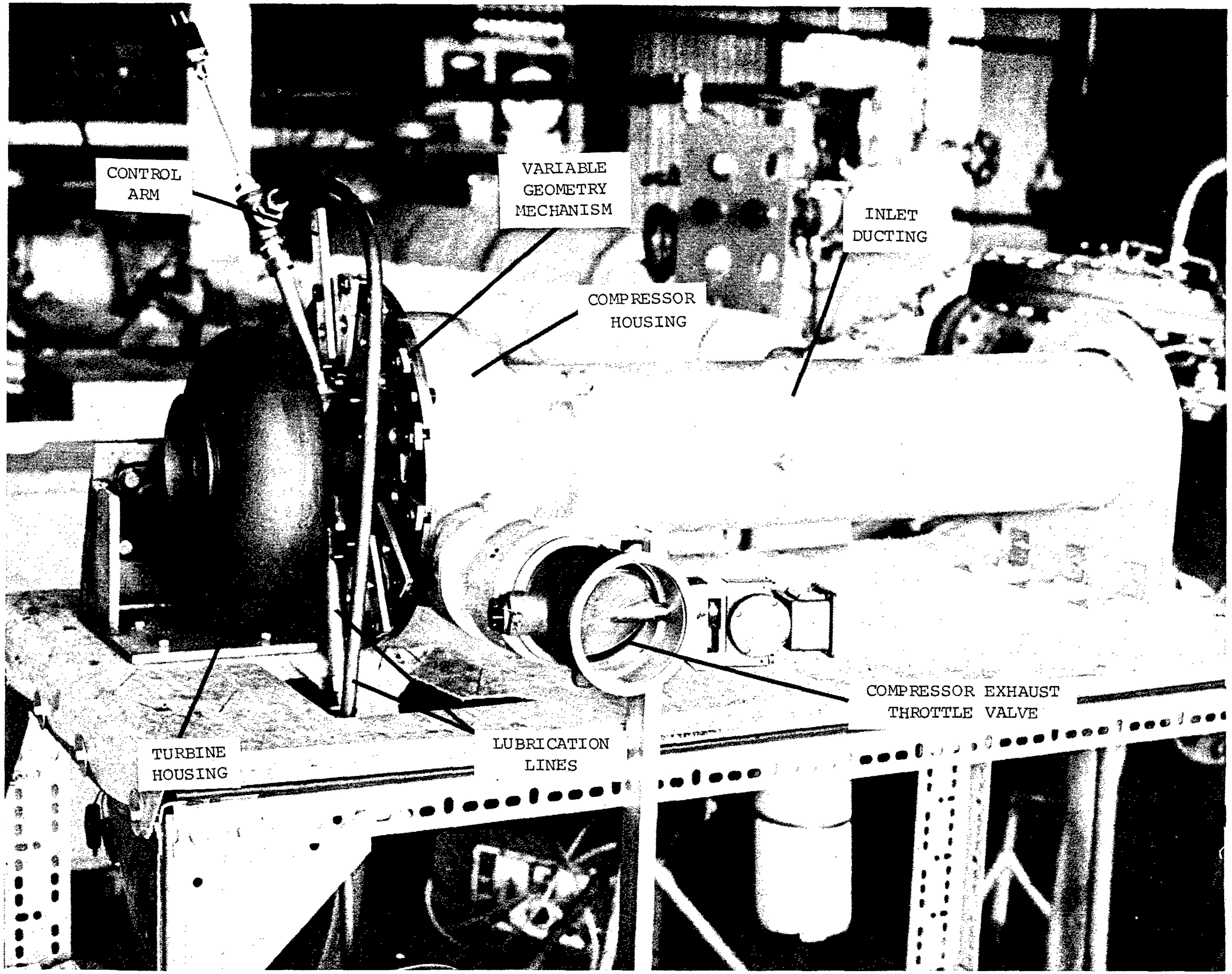


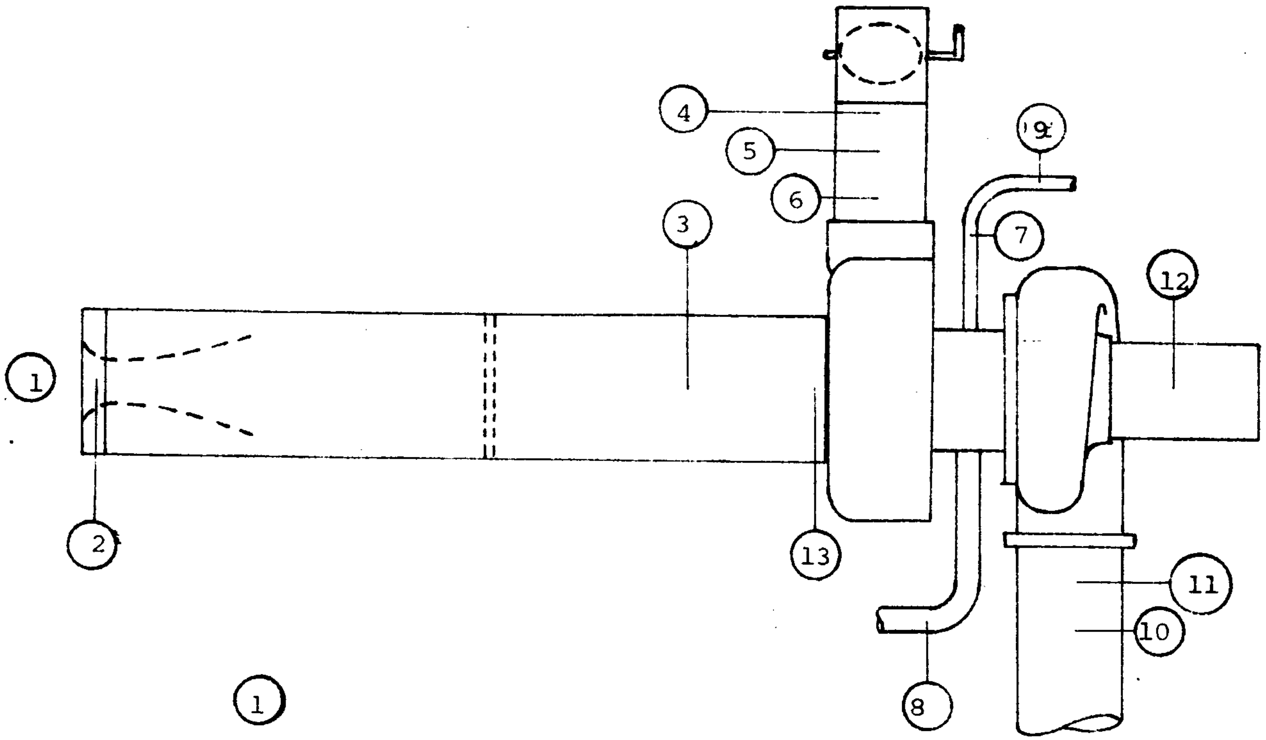
FIGURE 4.3.1. Variable Diffuser Compressor Test Rig.

TABLE III

VARIABLE DIFFUSER COMPRESSOR

TEST RIG INSTRUMENTATION LIST

ITEM	FUNCTION	SENSOR	QTY	RANGE	READOUT
1.	Ambient Temperature	Thermometer	2	0 - 120°F	Visual
2.	Venturi Throat ΔP	Static Tap	1	0 - 60 in.H_2O	Manometer
3.	Compressor Inlet Pressure	Static Tap	3	0 - 60 in.H_2O	Manometers
4.	Compressor Discharge Pressure	Static Tap	2	0 - 75 psia	Kollsmans
5.	Comp. Discharge Total Pressure	Probes	2	0 - 75 psia	Kollsmans
6.	Comp. Discharge Temperature	Thermocouple	2	40 - 450°F	Leeds-Northrup Bridge
7.	Oil Temperature (IN)	Thermocouple	1	40 - 250°F	Brown Recorder
8.	Oil Temperature (OUT)	Thermocouple	1	40 - 300°F	Brown Recorder
9.	Oil Pressure	----------	1	0 - 50 psig	Bourdon Gauge
10.	Turbine Inlet Pressure	Static Tap	2	0 - 80 psia	Kollsmans
11.	Turbine Inlet Temperature	Thermocouple	2	40 - 1000°F	Leeds-Northrup Bridge
12.	Turbine Exit Temperature	Thermocouple	2	0 - 700°F	Leeds-Northrup Bridge
13.	Speed	Magnetic Pick-up	1	0 - 99,999 RPM	Digital Counter
14.	Facility Orifice Differential Air Pressure for Air Flow	Static Tap	2	0 - 100 psia	Bourdon Gauge
15.	Facility Air Temperature	Thermocouple	2	40 - 1000°F	Leeds-Northrup Bridge



1. Ambient Temperature (2).
2. Venturi Throat ΔP (1).
3. Compressor Inlet Pressure (3).
4. Static Compressor Discharge Pressure (2).
5. Total Compressor Discharge Pressure (2).
6. Compressor Discharge Temperature (2).
7. Oil Inlet Temperature (1).
8. Oil Outlet Temperature (1).
9. Oil Inlet Pressure (1).
10. Turbine Inlet Pressure (2).
11. Turbine Inlet Temperature (2).
12. Turbine Exit Temperature (2).
13. Shaft Speed.

FIGURE 4.3.2 Variable Diffuser Compressor Test Rig Instrumentation Location.

Engineering Test Laboratories (AETL), Chatsworth, Calif. Heated compressed facility air was available to drive the compressor with the standard turbocharger radial turbine. Control of compressor speed was maintained manually with an electrically controlled throttling valve on the facility air supply line.

Initial pretest checks were made at low compressor speeds (roughly 10%) to determine that the mechanical operation of the rig was normal and that all instrumentation was functioning. Compressor performance was mapped by setting a predetermined corrected speed and diffuser vane throat area position and gradually increasing the back pressure with the discharge throttle valve. Enough steady state data points were taken at each speed to completely define the shape of the speed line characteristic between surge and choke. The approximate time between data points was 5 minutes. Surge was determined audibly.

The diffuser vane settings selected for testing were the 45%, 72%, 100%, and 110% of design diffuser throat area positions. Testing was planned at corrected speeds corresponding to those provided on the standard compressor map. These speeds included 28,700, 47,100, 56,500 and the design speed of 70,800 RPM. The order of testing was planned such that lower speeds were tested initially at the various diffuser settings, gradually working up in speed so that performance at maximum speed was obtained last. This was done so that in case of a failure at rated speed the greatest amount of data would already have been obtained. Also, actual hard surge was avoided at the rated speed of 70,800 RPM due to the possibility of damage to the rig, but an attempt was made to gain close proximity to the surge limit. Testing at lower speeds of 28,700 and 47,100 RPM was not planned at the 110% diffuser throat area setting because this regime of operation is not likely to be utilized in matching the variable diffuser compressor to the diesel engine.

4.4 Test Results and Performance Summary

Actual testing of the variable diffuser compressor commenced on 30 April 1973. The rig exhibited smooth operation throughout all of the part speed tests. At each part speed tested (roughly 40, 66 and 80% of design speed) surge was actually encountered and the surge airflow noted.

After completing the testing at 70,800 RPM with the diffuser in the 45% and 72% setting, a failure resulted while accelerating to the rated speed with the diffuser in the 100% (design) setting. Inspection revealed that a washer located next to the magnetized speed pickup nut on the compressor end of the shaft had failed, releasing a small pie-shaped portion of this washer (Figure 4.4.1) into the compressor inlet. Upon repeatedly striking inducer blades traveling at high speed, the failed washer damaged blade leading edges and the force of its impact apparently loosened the nut on the end of the shaft enough to allow the compressor and turbine wheels to rub against their respective housings. No apparent damage was sustained by the variable geometry diffuser system or the bearing housing.

At the time of failure the bulk of the proposed testing had been completed. The following chart illustrates the complete speed line data obtained before the failure resulted:

$N/\sqrt{\theta_{545}}$	45% Setting	72% Setting	100% Setting	110% Setting
28,700 (40.5%)	Yes	Yes	Yes	Testing Not Planned
47,100 (66.5%)	Yes	Yes	Yes	Testing Not Planned
56,500 (80%)	Yes	Yes	Yes	Yes
70,800	Yes	Yes	No	No

Only two speed lines had not been investigated of those which were originally planned. These were the 100% design speed at the design and 110% diffuser vane setting. The basic test performance parameters are tabulated in Table IV.

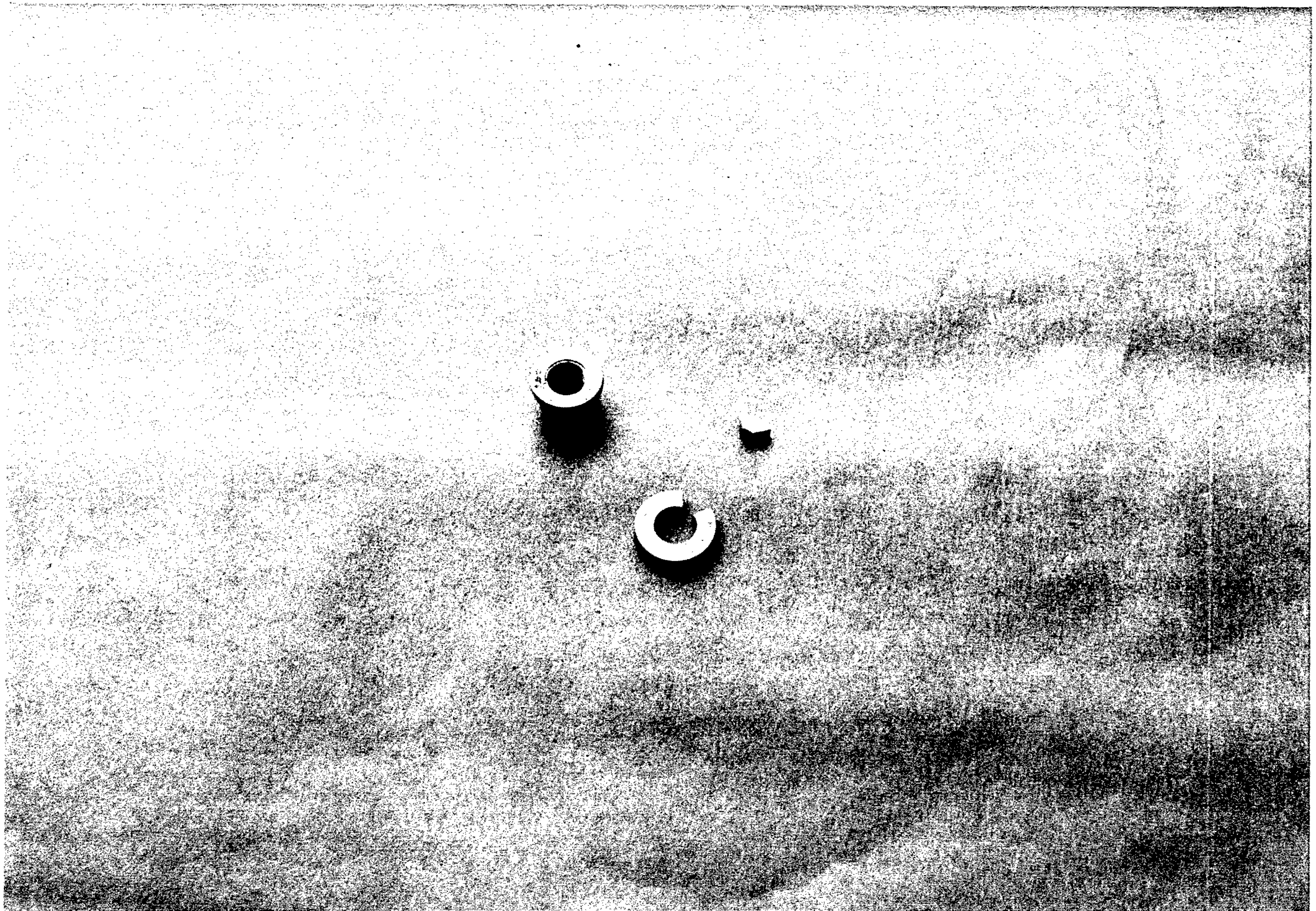


FIGURE 4.4.1. Failed spacer which caused erosive damage to inducer blade leading edges.

TABLE IV

JN BER	$\frac{W\sqrt{\theta_{545}}}{P_1/28.4}$	$\frac{N}{\sqrt{\theta_{545}}}$	T_1	T_2	P_1	P_2	$\left.\frac{P_2}{P_1}\right]_c$	η_c	VANE SETTING
			°R	°R	psia	psia			% A_D
001	32.50	27960	522	567.8	14.19	15.39	1.0916	.286	100
002	31.58	28313	522	569	14.19	16.33	1.1578	.471	100
003	28.30	27999	521	573	14.20	17.39	1.2303	.607	100
004	27.23	27999	520	572.5	14.20	17.74	1.2535	.655	100
005	26.08	28470	520	574.5	14.20	18.15	1.2838	.700	100
006	25.07	27696	519	572.4	14.20	17.99	1.2708	.682	100
007	21.11	27862	518	574	14.21	18.48	1.3047	.726	100
008	16.79	27783	518	577.2	14.22	18.81	1.3263	.731	100
009	13.97	27813	518	580	14.24	18.89	1.3298	.704	100
101	21.40	47088	537	708.2	14.25	17.28	1.2123	.175	45
102	20.52	47088	537	712	14.29	26.00	1.8194	.559	45
103	18.90	47088	537	715.3	14.33	26.80	1.8698	.575	45
104	16.18	47149	537	722.5	14.33	27.43	1.9138	.572	45
105	14.77	29041	536	614	14.28	17.13	1.1992	.367	45
106	12.98	29067	535	---	14.31	17.80	1.2439	----	45
107	12.97	29152	534	602.5	14.32	17.87	1.2483	.488	45
108	9.83	29152	534	607.4	14.33	18.68	1.3032	.545	45
109	8.09	29060	533	611	14.33	18.80	1.3119	.528	45
801	25.00	28674	543	595	14.10	14.49	1.0278	.1099	72
302	19.34	28420	543.5	601	14.18	17.90	1.2628	.6474	72
803	15.90	28706	544	604	14.20	18.46	1.2998	.7017	72
804	12.32	28693	544.5	603	14.27	18.61	1.3049	.7309	72
305	11.49	28680	545	610.5	14.28	18.65	1.3059	.6547	72
306	10.65	28667	545.5	613	14.30	18.69	1.3066	.6378	72
307	38.88	47280	545	702	14.20	20.00	1.4075	.4200	72
308	38.75	47280	545	704.5	14.20	24.15	1.6992	.5528	72
309	37.65	47121	544.5	708	14.22	27.51	1.9343	.6843	72
310	27.92	47241	544	711	14.23	28.00	1.9677	.6872	72
311	23.89	47323	544	717	14.24	28.25	1.9833	.6730	72
312	48.52	56751	544	765	14.17	24.91	1.7587	.4345	72
313	48.54	56871	544	765.5	14.16	31.01	2.1897	.6103	72
314	40.54	56811	544	773	14.19	35.83	2.5248	.7115	72
301	36.49	58458	528	774.3	14.23	37.83	2.660	.6846	72

TABLE IV - Continued

UN BER	$\frac{W\sqrt{\theta_{545}}}{P_1/28.4}$	$\frac{N}{\sqrt{\theta_{545}}}$	T_1	T_2	P_1	P_2	$\left.\frac{P_2}{P_1}\right]_c$	η_c	VANE SETTING
			^oR	^oR	psia	psia			% A_D
0902	35.37	56488	529.5	764.8	14.23	35.66	2.506	.6685	72
0903	32.48	56381	530.5	766.8	14.24	35.53	2.496	.6635	72
0904	29.60	56263	532	773.3	14.24	35.23	2.474	.6439	72
0905	26.90	56683	539.5	782.8	14.12	25.08	1.777	.3921	45
0906	26.82	56683	539.5	785.5	14.13	31.02	2.195	.5471	45
0907	25.11	56683	539.5	790.3	14.18	32.93	2.323	.5798	45
0908	20.94	56683	539.5	799.5	14.32	34.35	2.399	.5831	45
0909	19.91	56578	540.5	804.8	14.34	34.24	2.388	.5717	45
0910	50.83	47190	543	687	14.17	22.62	1.597	.5343	100
0911	47.13	47130	543	691.3	14.18	26.54	1.872	.7117	100
912	39.08	47220	545	698.5	14.25	28.10	1.973	.7527	100
913	34.67	47117	546	704.5	14.26	28.40	1.992	.7424	100
914	63.61	56758	548.5	764.5	14.10	24.95	1.770	.4451	100
915	62.90	56796	549	770.8	14.10	31.87	2.260	.6425	100
916	59.55	56648	549.5	772.5	14.11	34.28	2.429	.7035	100
917	53.21	56628	550	778.3	14.14	35.62	2.519	.7217	100
918	45.87	56597	550.5	788.3	14.17	36.42	2.570	.7107	100
919	69.10	56896	550.5	767.3	14.06	29.53	2.102	.5943	110
920	67.23	56896	550.5	771.3	14.08	33.54	2.383	.6958	110
921	58.81	56776	550.5	775	14.12	35.64	2.524	.7356	110
922	50.34	56896	550.5	784.5	14.15	36.46	2.576	.7222	110
923	39.24	71104	550.5	925	14.19	38.48	2.712	.4799	45
01	40.28	70791	535	909.3	14.23	39.60	2.796	.4818	45
02	40.16	70731	535	909	14.23	45.42	3.156	.5486	45
03	39.26	70791	535	914.5	14.24	47.59	3.342	.5731	45
04	38.09	71215	535	913	14.24	48.53	3.408	.5867	45
05	35.79	71215	535	916.8	14.25	50.73	3.561	.6061	45
06	66.42	70840	535	881	14.12	41.75	2.936	.5421	72
07	66.42	70886	535	882.8	14.14	46.75	3.311	.6200	72
08	64.89	70791	535	886.5	14.13	50.66	3.585	.6621	72
09	57.80	70734	535	896	14.16	53.66	3.789	.6788	72
10	61.19	70613	535	892.3	14.16	52.26	3.692	.6678	72

In general, the performance of the variable geometry compressor was very close to the predicted levels of Figure 4.1.3, which were based on the results of tests of similar variable diffuser systems. Figure 4.4.2 illustrates the effect of diffuser vane setting on flow range and pressure ratio for the test compressor. Choked mass flow at any speed was found to be approximately proportional to the diffuser throat area. The surge flow deviated from this trend somewhat, especially at a corrected speed of 56,500 RPM, where the speed line characteristics at both the 45% and 72% throat area settings exhibited extra surge range. Increased stability at 56,500 RPM was especially obvious at the 72% diffuser setting. The reason for this is not known, but the result is obviously beneficial. At a pressure ratio of 2.4 the surge range was 3.5:1 between throat settings of 110% and 45%. A surge range of 2.5 was obtained at a pressure ratio of 3.4 between throat settings of 100% and 45%. With the vanes positioned in the 100% area setting the test compressor essentially matched the pressure rise and flow rate of the production compressor (shown dashed on Figure 4.4.2). Even during testing at the lowest throat area setting (45%) there was no indication that flow reduction control with the variable diffuser was affected by inducer stalling phenomenon; that is, even though the inducer exhibited stalling the diffuser continued to control the flow rate of the machine. Based upon previous tests (at Solar, Ref. 5) it is expected that effective flow control could have been maintained down to perhaps a 16% throat setting, although certainly at substantially lower efficiency levels.

The stage total pressure ratio was measured from total pressure probes in the exhaust duct and wall static probes in the inlet duct. The inlet duct was sufficiently large that for practical purposes the total and static pressures were equal. This inlet pressure measurement technique eliminated the necessity of installing total pressure probes in the inducer inlet where wakes could cause problems. In tests, the maximum pressure ratio obtained at a given rotor speed decreased as the diffuser throat area was reduced below the design value. This was

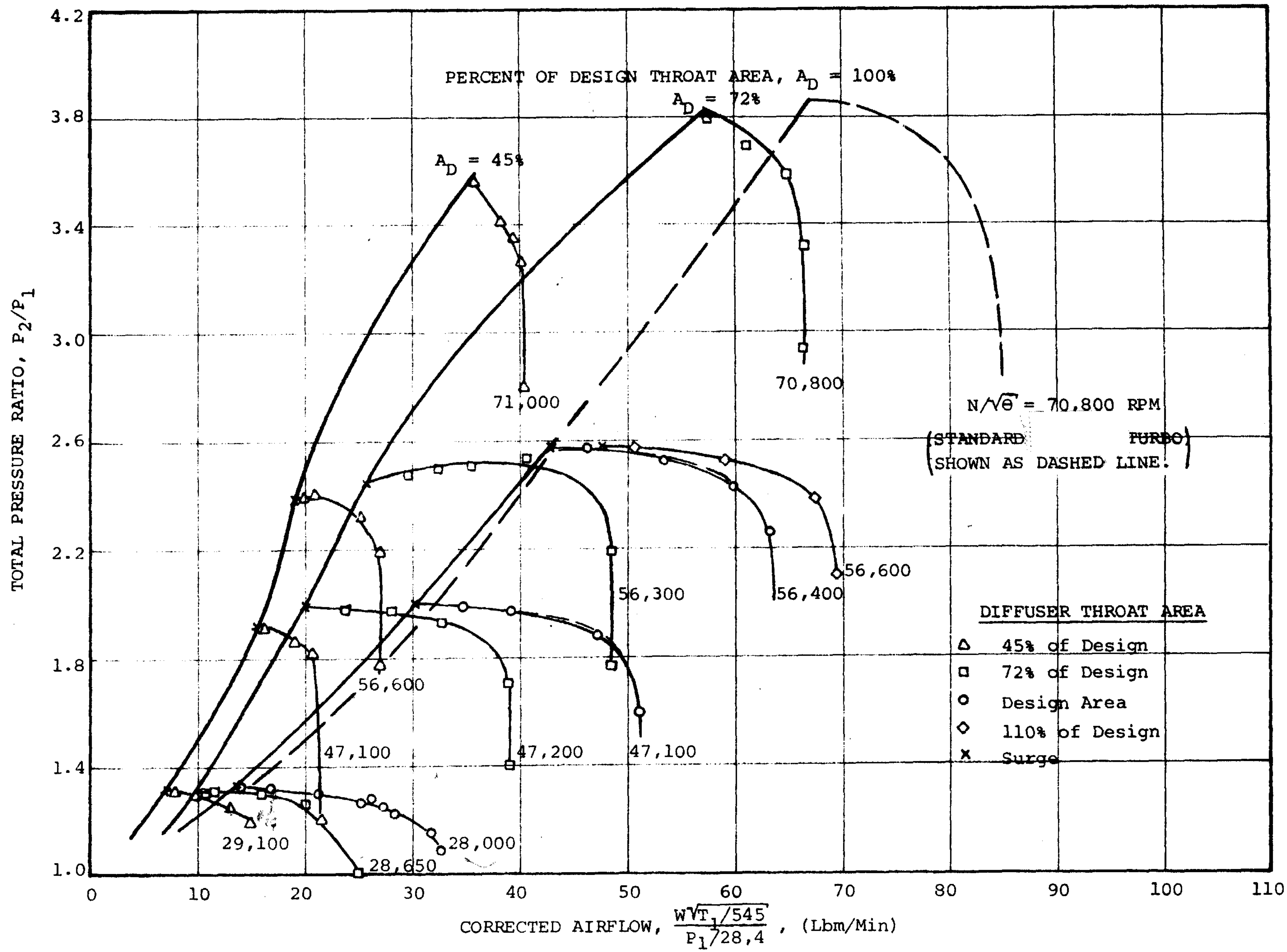


FIGURE 4.4.2. Effect of Vane Setting on Surge Characteristics and Pressure Ratio of the Variable Geometry Test Compressor.

caused by a reduction in compressor efficiency at reduced flow rates and is shown clearly on the efficiency maps of Figure 4.4.3, 4.4.4, and 4.4.5.

Compressor efficiency was evaluated from measured temperature rise and inlet to discharge total pressure rise. Comparing Figure 4.4.4 and 4.4.5, a small efficiency penalty was incurred at the 72% setting (approximately 4 points) relative to the design setting. This loss represents roughly one-half of that experienced in the Solar variable geometry tests. Accordingly, the efficiency obtained exceeded the predicted peak efficiency of 66 percent at this vane setting (Figure 4.1.3).

Part of the improvement relative to the Solar results at this vane setting can be explained by the use of offset of the diffuser vanes on the pivot shaft (which in the TMSCO design allows the radial gap between the impeller tip and the vane leading edge to be kept relatively constant as the vanes are rotated to closed throat area settings). Even small increases in this so called vaneless space diameter ratio have a large effect on frictional flow path length, and therefore cause reduced efficiency.

With the diffuser throat area reduced to 45% the maximum efficiency was 59 percent, a reduction of 15 points relative to the design setting. This loss is roughly equal to the value predicted from the Solar results (58 percent) at the same vane setting.

At the low diffuser throat area (45%) setting a slightly higher than ambient wall static pressure was measured in the compressor inlet duct while testing near the surge line. This phenomenon is believed to be caused by recirculation of air from the impeller back into the inlet duct (caused by a heavily stalled impeller), which induces a vortex or prewhirl in the direction of rotation of the rotor. The potential pressure rise of the stage and thus the efficiency is adversely affected by not only the recirculation of heated air into the inlet but also the induced prewhirl. In the Solar variable geometry tests a similar effect was also observed, even though the inlet design in those

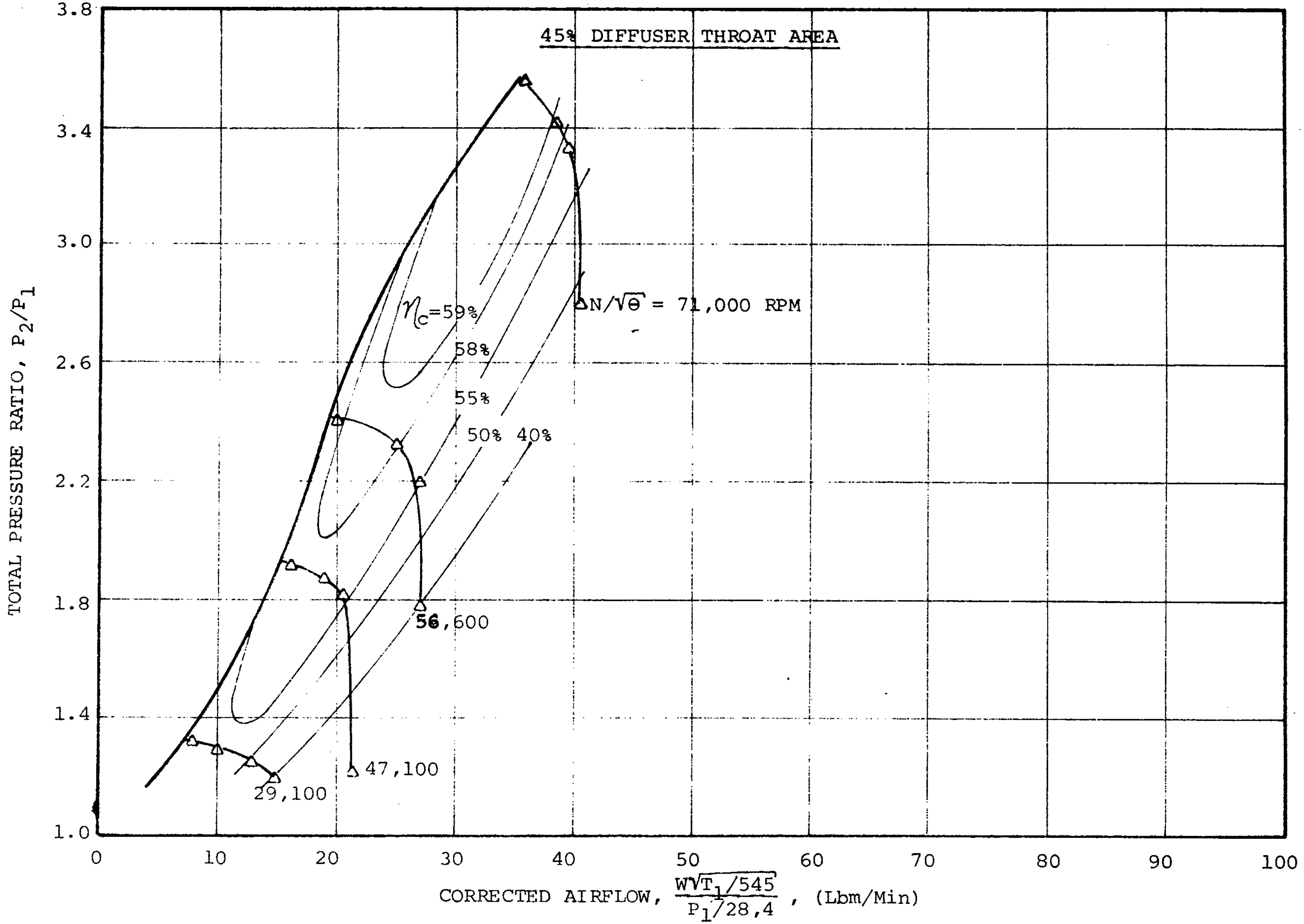


FIGURE 4.4.3. Performance of Variable Geometry Compressor in 45% Vane Setting.

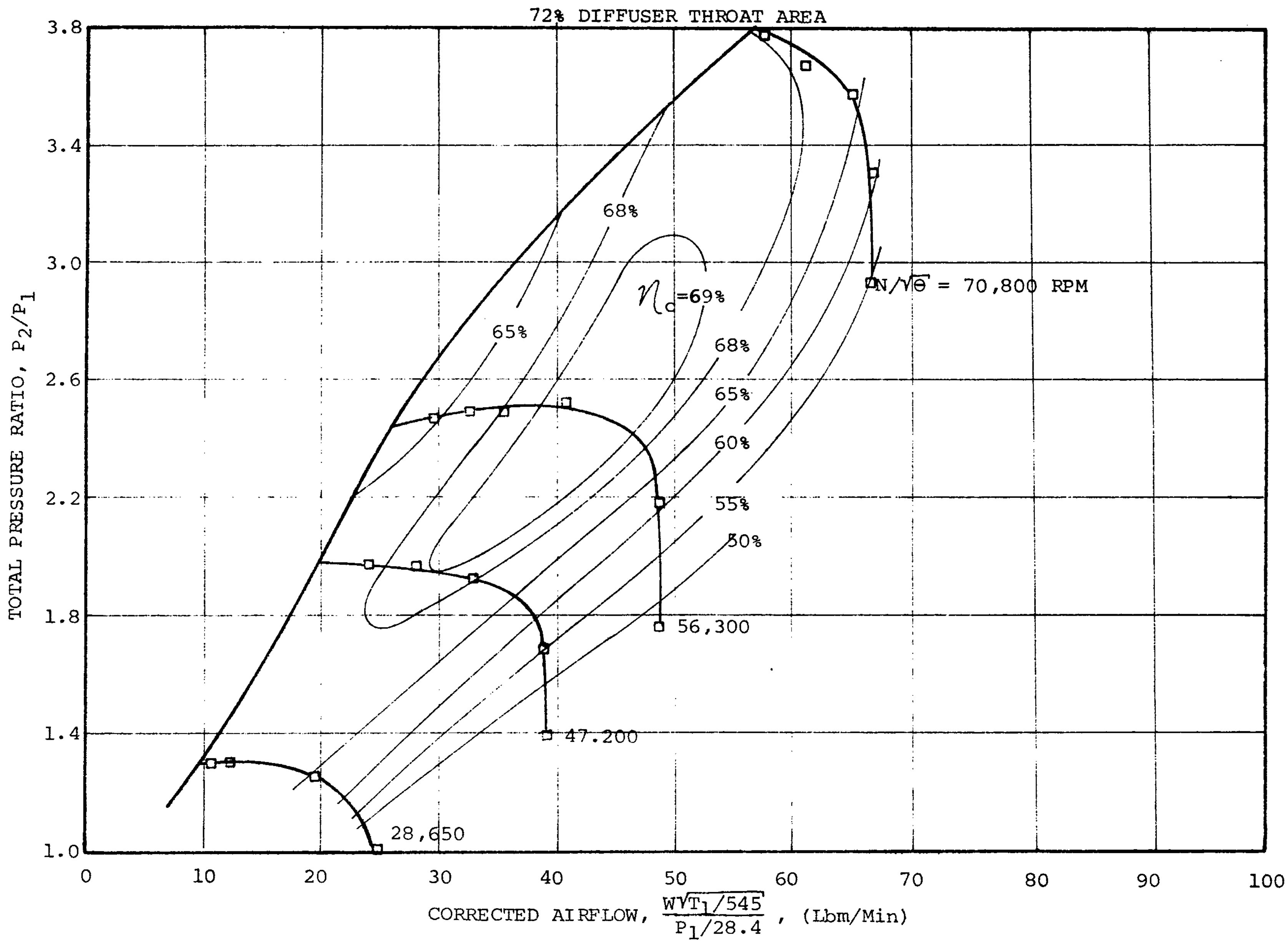


FIGURE 4.4.4. Performance of Variable Geometry Compressor in 72% Vane Setting.

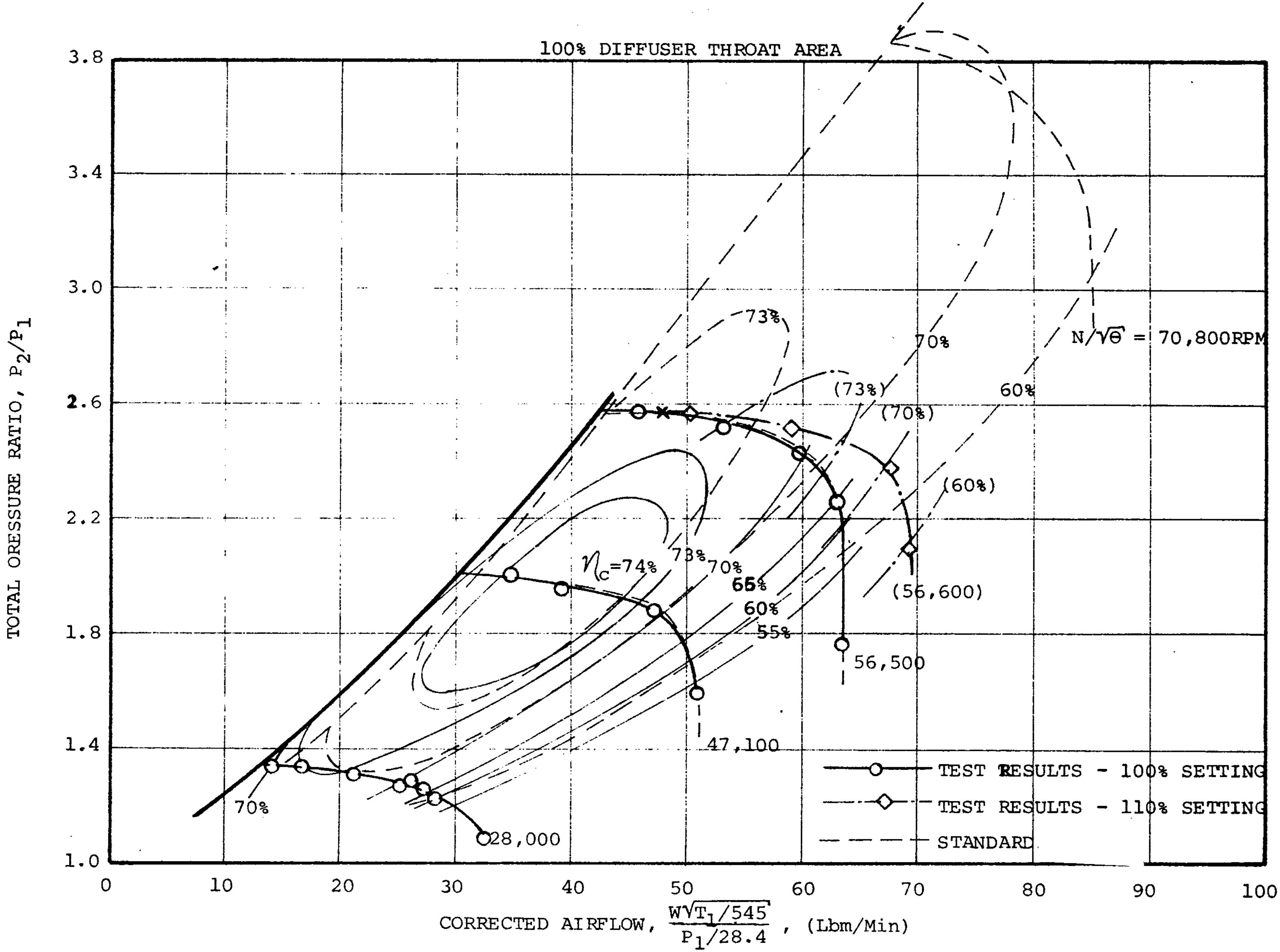


FIGURE 4.4.5. Performance of Variable Geometry Compressor in 100% and 110% Vane Settings.

tests tended to restrict the formation of a vortex because large vane-type struts were used to support the stationary hub cone. It is probable that the restriction of the inlet vortex in the Solar tests reduced the efficiency losses at low flow rates and is the primary reason why the TMSCO data did not show the relative improvement in efficiency at the 45% setting that it exhibited at 72% setting. In the future, it may be possible to devise effective means to restrict the recirculation of flow at very low diffuser vane settings, thus improving the efficiency and useful flow range of variable diffuser compressors. This is an area for future development work in variable geometry compressor systems.

At the design setting the experimental data closely matched the quoted performance of the standard compressor. In Figure 4.4.5 a comparison of efficiency of the standard compressor and the test compressor is presented. The efficiency islands are almost coincident, and pressure ratio at a given speed showed very close agreement with the standard compressor map, so it can be concluded that the variable geometry mechanism had virtually no effect upon performance of the compressor in the design diffuser configuration.

A single 80% speed line was run at 110% vane setting to demonstrate the feasibility of improved matching of the compressor and engine at near rated speed and boost levels. Under these engine conditions the operating point is typically located far from the surge line, whereas maximum efficiency in radial-bladed compressor is usually higher near the surge line. By opening the vanes to the 110% setting it was found that higher efficiency was obtained at higher flow rates as expected. In addition, a slight increase (about one percent higher) measured compressor efficiency was obtained at this setting relative to the design setting. The surge range, defined as the percentage of surge to choke flow, was also improved slightly.

The performance obtained at the 110% vane setting demonstrated that high efficiency operation is possible at larger than design throat area. As is true with most compressors optimized for good surge range, the design diffuser throat area of this compressor was smaller than the geometry which gives maximum efficiency. The variable geometry version of this compressor is actually oversized for the VHO engine application because at least 10% more airflow than the VHO would require can be handled without efficiency penalty compared to the fixed geometry version. Proper matching of a variable diffuser compressor to a given engine application requires that the full range of high efficiency diffuser throat area settings be utilized from the high flow rate at rated power down to the most critical surge margin condition at lower speeds. In these tests, a surge range of 2.6:1 was obtained between throat settings of 110% and 72% at a pressure ratio of 2.4:1. The difference in compressor efficiency over this broad range of airflows was less than 5%.

5.0 CONCLUSIONS

Based on the results of the work reported herein, the following conclusions are drawn:

1. A variable diffuser is the most practical method of obtaining the required degree of surge control projected for advanced diesel turbocharger compressors.
2. Performance data was obtained for the variable diffuser compressor at diffuser throat areas between 45 and 110 percent of the design setting over a range of corrected speeds from 40 to 100 percent of design (70,800 RPM). Results indicated that between the 110 and 45 percent vane settings approximately 3.5:1 flow range was achieved without excessive efficiency penalty at a pressure ratio of 2.4:1. Only 5 points difference in compressor efficiency was observed between throat settings of 110 and 72 percent of design (corresponding to surge range of 2.6:1) at this same pressure ratio. At a pressure ratio of 3.4:1 a surge range of 2.5:1 was obtained between throat settings of 100 and 45 percent of design. Performance of the compressor in the design setting exactly matched the predicted performance of the production compressor, so there was no effect of variable geometry on peak efficiency potential. Testing demonstrated that the variable diffuser actuator system is feasible and that by utilizing proper design techniques a high level of efficiency is obtained over the broadest possible range of flow rates and pressure ratios.
3. Based on the results of these tests, it is advisable to size a variable diffuser compressor to operate over a

range of throat settings from a maximum value (corresponding to a setting greater than that which provides maximum efficiency but slightly less than that which results in substantial inefficiency due to inducer choking) down to a minimum value required to handle the most critical surge margin requirement of the engine. The reason for matching in this manner is to obtain maximum compressor efficiency at the most critical boost horsepower requirement, probably a medium engine speed, high pressure ratio (maximum engine torque) condition.

6.0 LIST OF SYMBOLS AND ABBRIEVIATIONS

A	Area, in^2
A_D	Diffuser throat area, in^2.
BSFC	Brake specific fuel consumption, $\frac{lb/hr.}{BHP}$
BHP	Brake horsepower
CFM	Volumetric flow rate, cu.ft./min.
d	Diameter, in.
E	Euler work, ft.lbf/lbm
f/a	Fuel-air ratio
g_c	Gravitational constant, 32.2 ft/sec^2
H_{ad}	Adiabatic head, $JC_p \triangle T_{ad}$, ft.
HP	Horsepower
IGV	Inlet guide vane
J	Joules equivalent of heat, 778 ft.lbf./Btu
L	Length
M	Mach number
N	Rotational speed, rev./min.
N_s	Specific speed, $rpm \cdot cfs^{0.5}/H_{ad}^{0.75}$
P	Absolute total pressure, psia, also inches of Hg
p	Absolute static pressure, psia, also inches of Hg
RMS	Root mean square
T	Total temperature, ^0R
V_u	Absolute tangential velocity, ft./sec.
V	Volumetric flow rate, cfm

W	Airflow, lb/min., also
	Relative Velocity, ft/sec.
U	Blade tip speed, ft./sec.
ϵ	Impeller diameter ratio, tip diameter/inducer RMS diameter
γ	Specific Heat ratio, 1.395
Δ	Difference
$\delta_{28.4}$	Relative inlet absolute pressure, $\frac{P_1, \text{ in. Hg}}{28.4}$
θ_{545}	Relative inlet temperature, $\frac{T_1, {}^{\circ}R}{545}$
θ_1	Inlet prewhirl, degrees
π	Pi, 3.1416
ϕ_2	Flow parameter, $\frac{4 V_2}{\pi D_2^2 U_2}$ dimensionless
η	Compressor adiabatic efficiency, $\frac{(P_2/P_1)^{\frac{\gamma - 1}{\gamma}} - 1}{T_2/T_1 - 1}$
ψ	Head coefficient, $\frac{\Delta H_{ad}}{U_2^2/2g}$, dimensionless
β	Blade angle (relative to axial direction), deg

SUBSCRIPTS

1	Impeller inlet
2	Compressor exit, also impeller exit
C	Compressor
D	Diffuser
m	Meridional direction
u	Tangential direction
x	Axial direction

7.0 REFERENCES

1. Shouman, A. R. and Anderson, J. R.; "The Use of Compressor-Inlet Prewhirl for the Control of Small Gas Turbines", Trans. ASME, Series A, Vol. 86, 1964, pp. 136-140.
2. Shouman, A. R.; "Prewhirl an Added Degree of Freedom to the Designer of Small Single-Shaft Gas Turbines", ASME 66-6T-89.
3. Stepanoff, A. J.; "Inlet Guide Vane Performance of Centrifugal Blowers", Trans. ASME, Series A, Vol. 83, pp. 371-380.
4. Rodgers, C. and Sapiro, L.; "Design Considerations for High-Pressure Ratio Centrifugal Compressors", ASME 72-GT-91, San Francisco, Calif., March 1972.
5. Rodgers, C.; "Variable Geometry Gas Turbine Radial Compressor" ASME 68-6T-63, Washington, D.C., March 1968.
6. Loch, Dr. E.; "Single Stage Turbo-Refrigerators", Escher Wyss News, date unknown.
7. Pampreen, R. C.; "The Use of Cascade Technology in Centrifugal Compressor Vaned Diffuser Design", ASME 72-GT-39, San Francisco, Calif., March 1972.
8. Rodgers, C.; "Typical Performance of Gas Turbine Radial Compressors" Trans. ASME, Ser. A., Vol. 86, pp. 161-175.
9. Laskin, E. B. and Kofskey, M. G.; "Increase in Stable Air Flow Operating Range of a Mixed Flow Compressor by Means of a Surge Inhibitor", NACA RM E7C05, 1947.
10. Speer, I. E. "Design Considerations and Development of a Broad Range, High Efficiency Centrifugal Compressor for a Small Gas Turbine". Trans. ASME. Vol. 75, 1953.
11. Rodgers, C. and Mnew, H.; "Rotating Vaneless Diffuser Study", Final Report to U. S. Army Mobility Equipment Research and Development Center, Ft. Belvoir, Va.; AD 716370, Oct. 1970.

12. Timoney, S. G.; "Diesel Design in Turbocharging", paper printed in SAE Journal, April 1966.

13. Schweitzer, P. H. and Tsu, T. C.; "Energy in the Engine Exhaust", Trans. of ASME, 1949, pp. 665-672.

14. Kline, S. J., Renean, L. R. and Johnston, J.P.; "Performance and Design of Straight Two-Dimensional Diffusers", Report PD-8, Stanford University, Stanford, Calif. (1962)

15. Shepherd, D. G.; Principles of Turbomachinery, The MacMillan Company, 1956.

16. Baumeister and Marks; Standard Handbook for Mechanical Engineers, McGraw-Hill, 1968

17. Lowy, R.; "Efficiency Analysis of Pelton Wheels", Trans. of ASME, 1944.

18. Smith, V. J.; "A Review of the Design Practice and Technology of Radial Compressor Diffusers", ASME Paper 70-GT-116.

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13. ABSTRACT

An analytical and experimental investigation was conducted to determine the most practical method of surge control for advanced diesel turbocharger compressors. Results of the analysis indicated that a vaned diffuser actuation system represented the best method of achieving effective regulation over a broad range of flow rates. Bench testing of the fabricated hardware indicated that the variable diffuser method has the ability to regulate flow and control surge without excessive reduction of efficiency or pressure ratio.

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14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Diesel Engines (Components)						
Turbocharger						
Surge Control (Compressor)						
Compressors; Variable Geometry						
Diesel Engine Smoke Reduction						
Diesel Engine - Low Speed Power Increase						